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F16H 45/02 61/14

(52) UK CL (Edition P)
F2C C1C16 C1C3B3 C1C4B14 C1C4B3 C1C9B1 C1E2 C1E6

(56) Documents Cited
GB 2280733 A GB 2275513 A GB 2127916 A

(58) Field of Search
UK CL (Edition P) F2C
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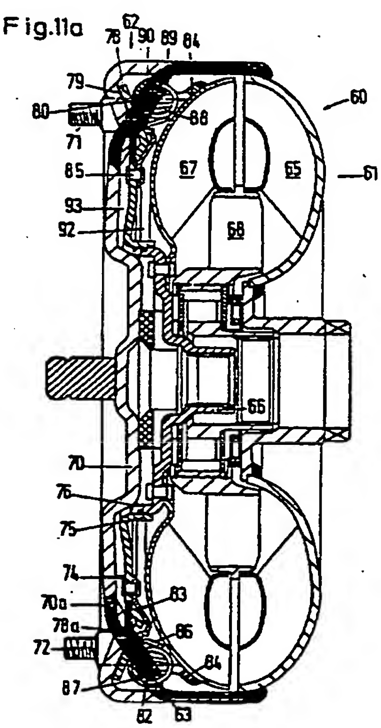
(54) Lock-up clutch for a torque converter

(57) A lock-up clutch for a hydrodynamic torque converter of a torque transfer system has a pump wheel 17, a turbine wheel 18, a guide wheel 19 and a converter cover 16. The cover is centred on the axis of rotation, is connected rotationally secured with the pump wheel and encloses the turbine wheel. A central ring piston which is mounted between the converter cover 16 and the turbine wheel 18 has its radially outer parts shaped to form a conical clutch friction disc 20. Radially inwards, the piston has a sealing hub 75 which is set on a counter sealing hub 76 which is connected rotationally fast with the turbine wheel.

Fig.11a



GB 2 320 531 A

Fig.1a

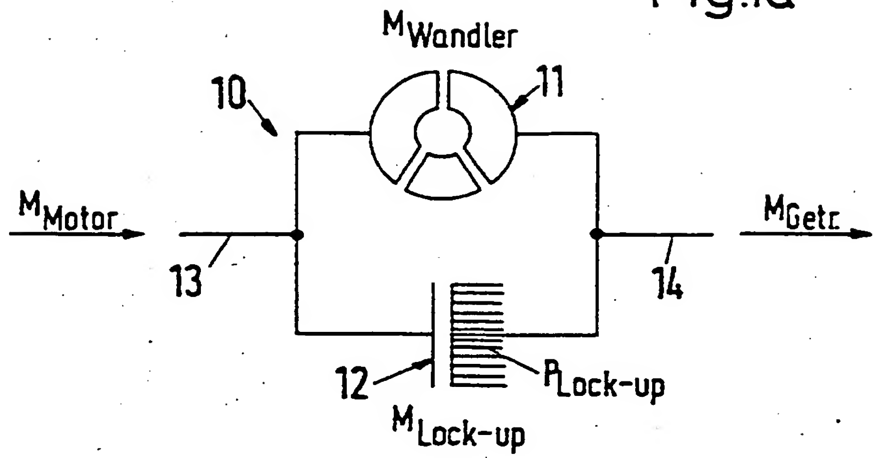

$M_{Wandler}$
10
11
M_{Motor}
$M_{Getr.}$
13
14
12
$P_{Lock-up}$
$M_{Lock-up}$

Fig.1b

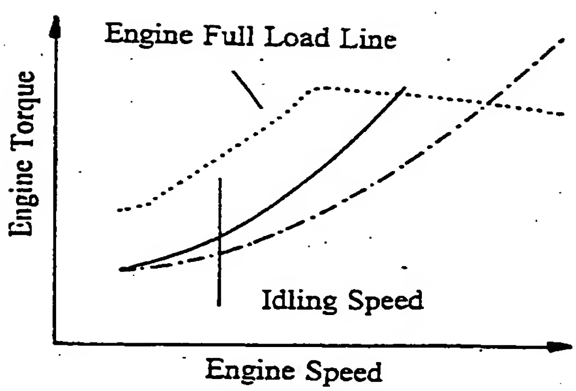

Engine Full Load Line
Engine Torque
Idling Speed
Engine Speed

Fig.1c

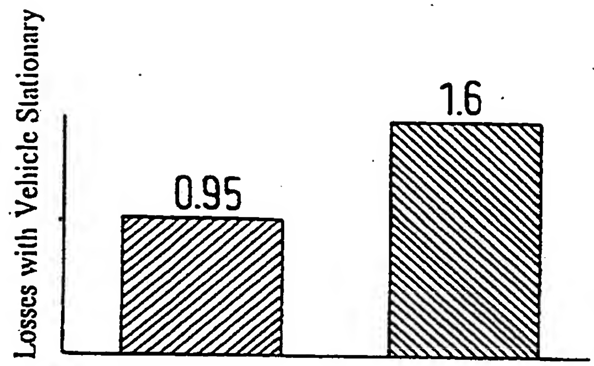
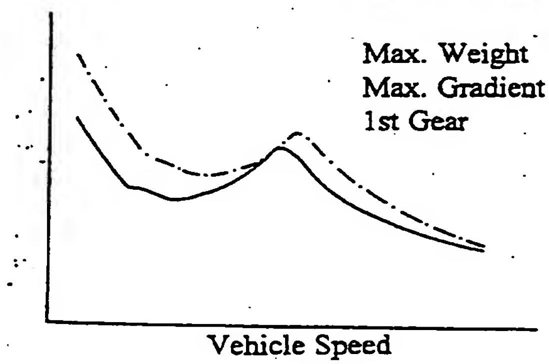
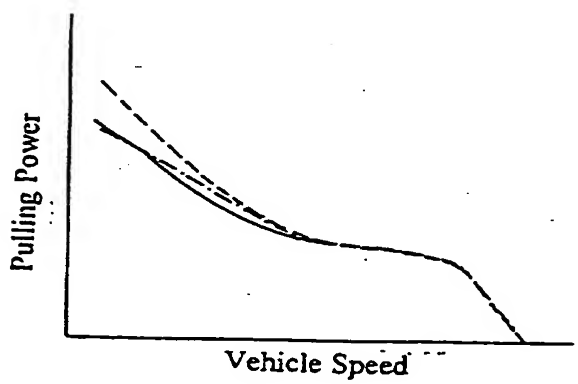


Fig.1d



— Conventional Converter — Soft Converter

Fig.1e



— Conventional Converter

--- Soft Converter

-- Soft Converter with Higher Conversion

Fig.2

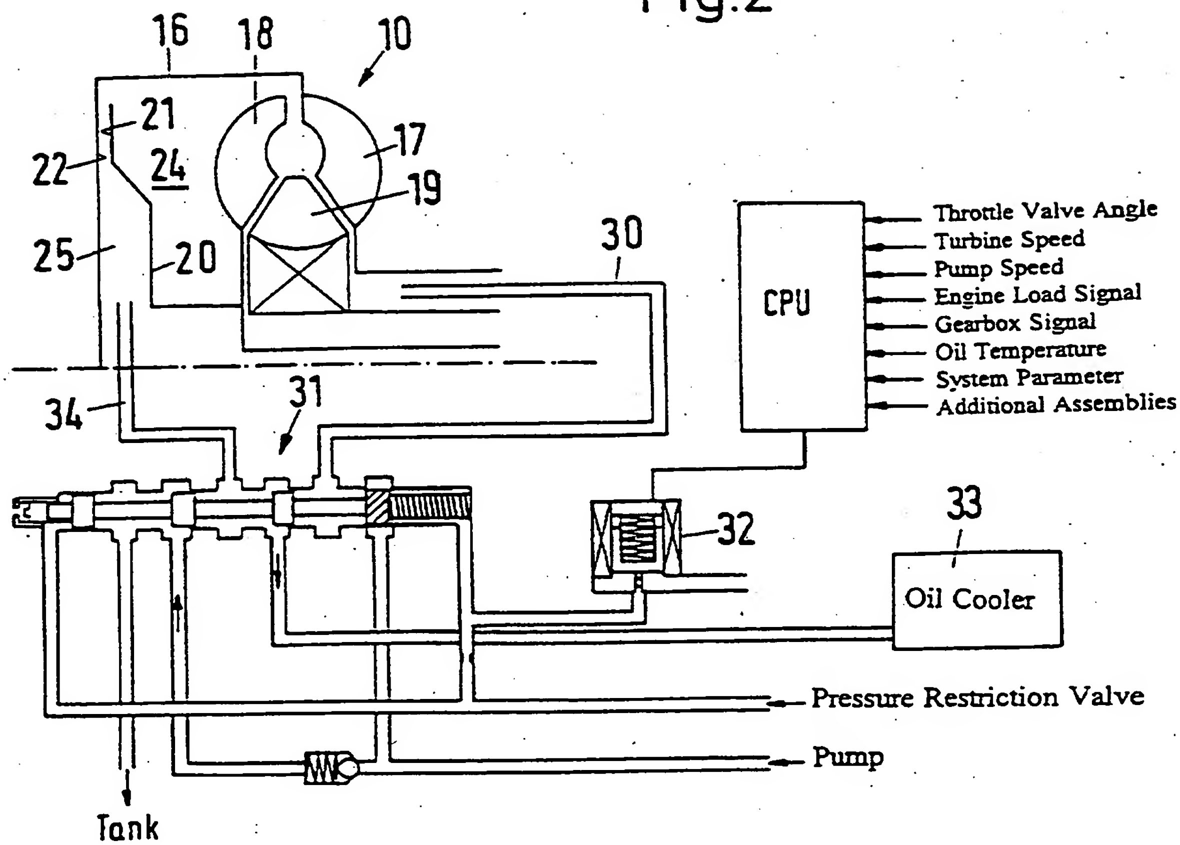

16
18
10
21
22
24
17
19
25
20
30
CPU
Throttle Valve Angle
Turbine Speed
Pump Speed
Engine Load Signal
Gearbox Signal
Oil Temperature
System Parameter
Additional Assemblies
34
31
32
33
Oil Cooler
Pressure Restriction Valve
Pump
Tank

Fig.3

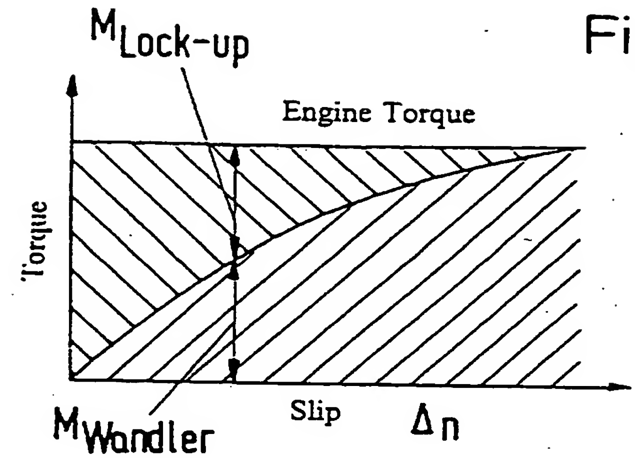


Fig.4

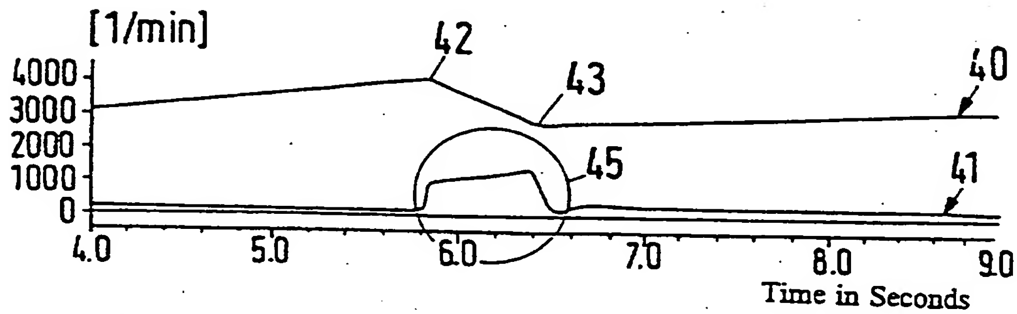


Fig.5

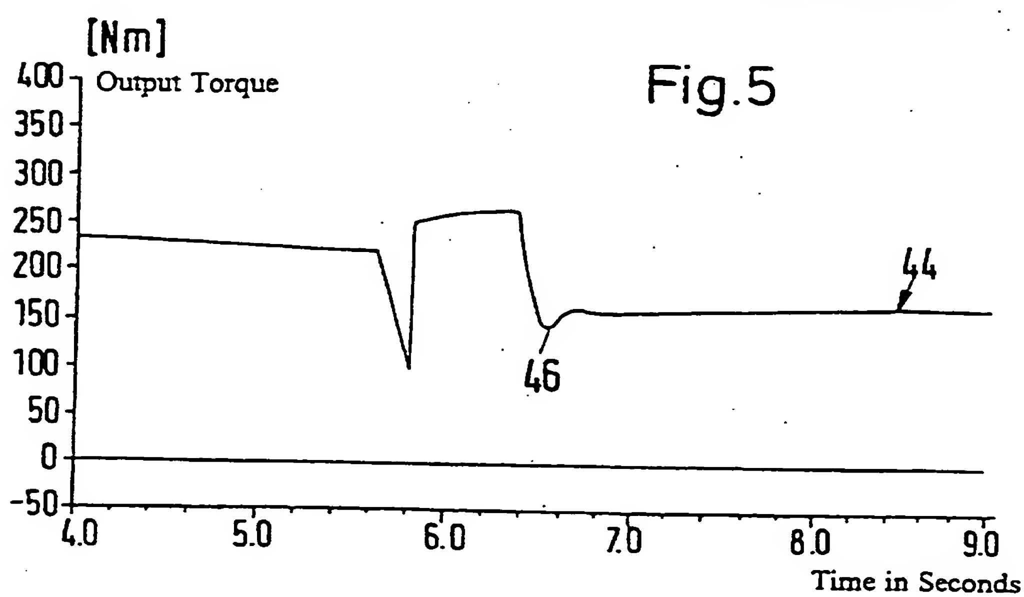


Fig.6

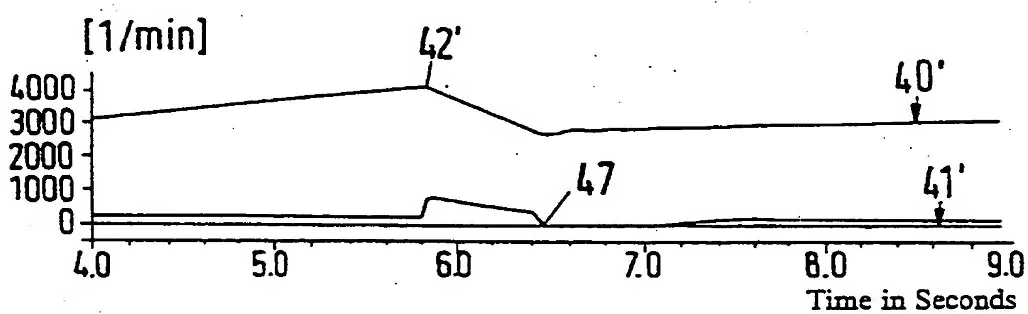

[1/min]
42'
40'
4000
3000
2000
1000
0
47
41'
4.0
5.0
6.0
7.0
8.0
9.0
Time in Seconds

Fig.7

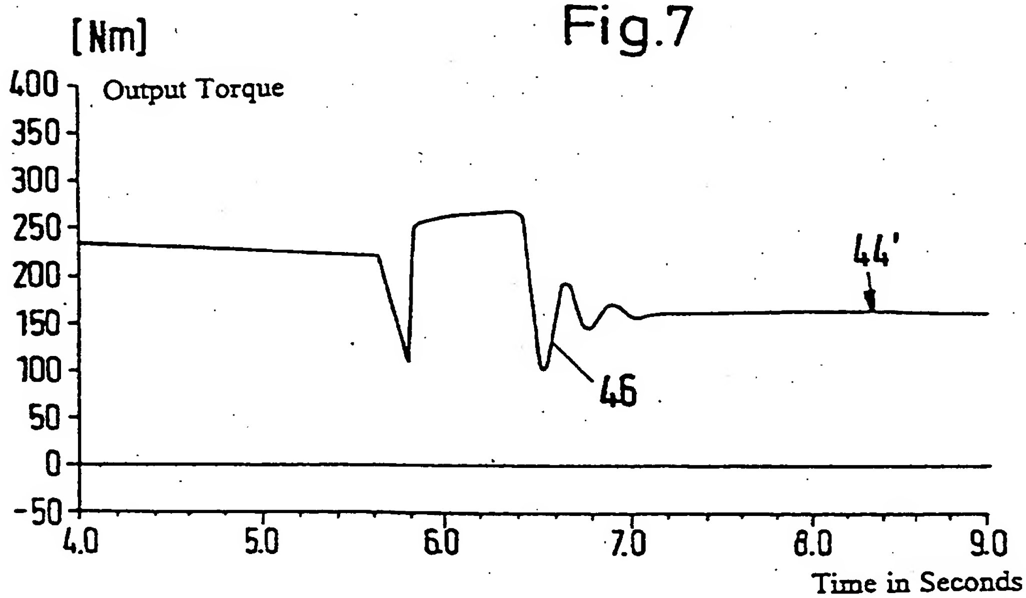

[Nm]
Output Torque
400
350
300
250
200
150
100
50
0
-50
44'
46
4.0
5.0
6.0
7.0
8.0
9.0
Time in Seconds

Fig.8

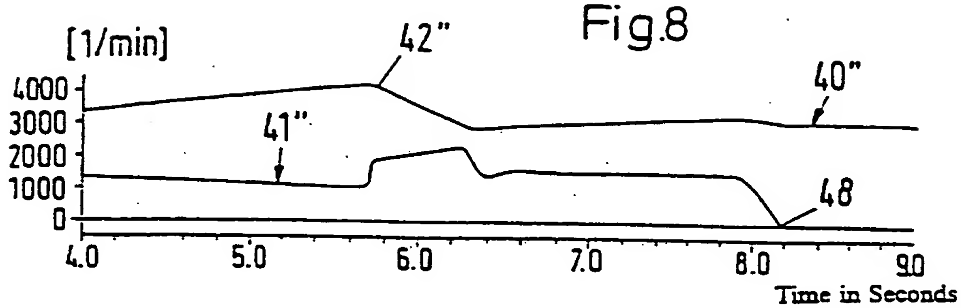


Fig.9

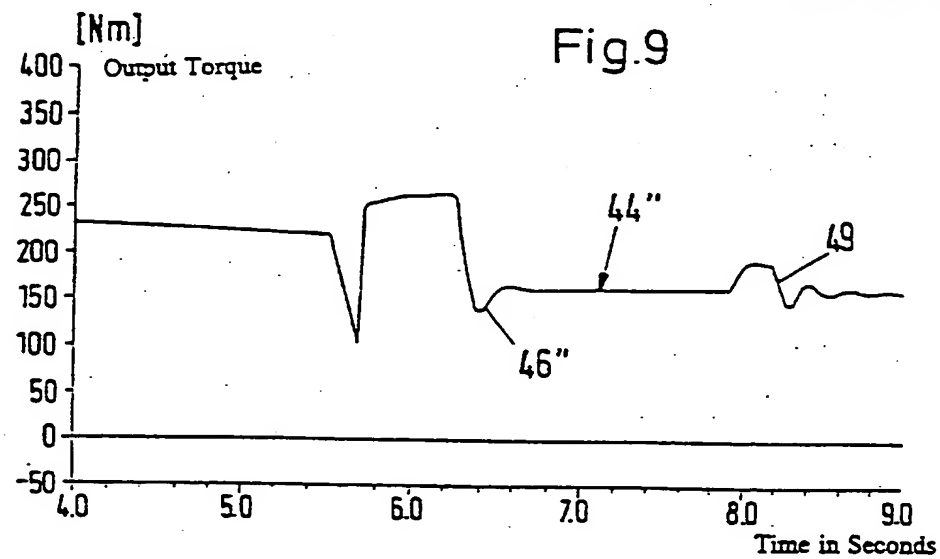


Fig.10

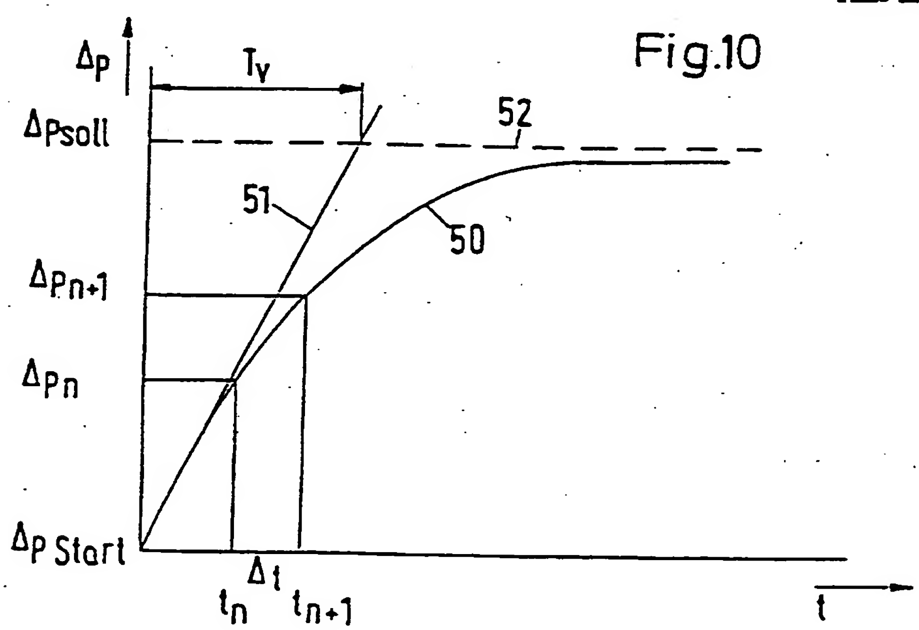


Fig.11a

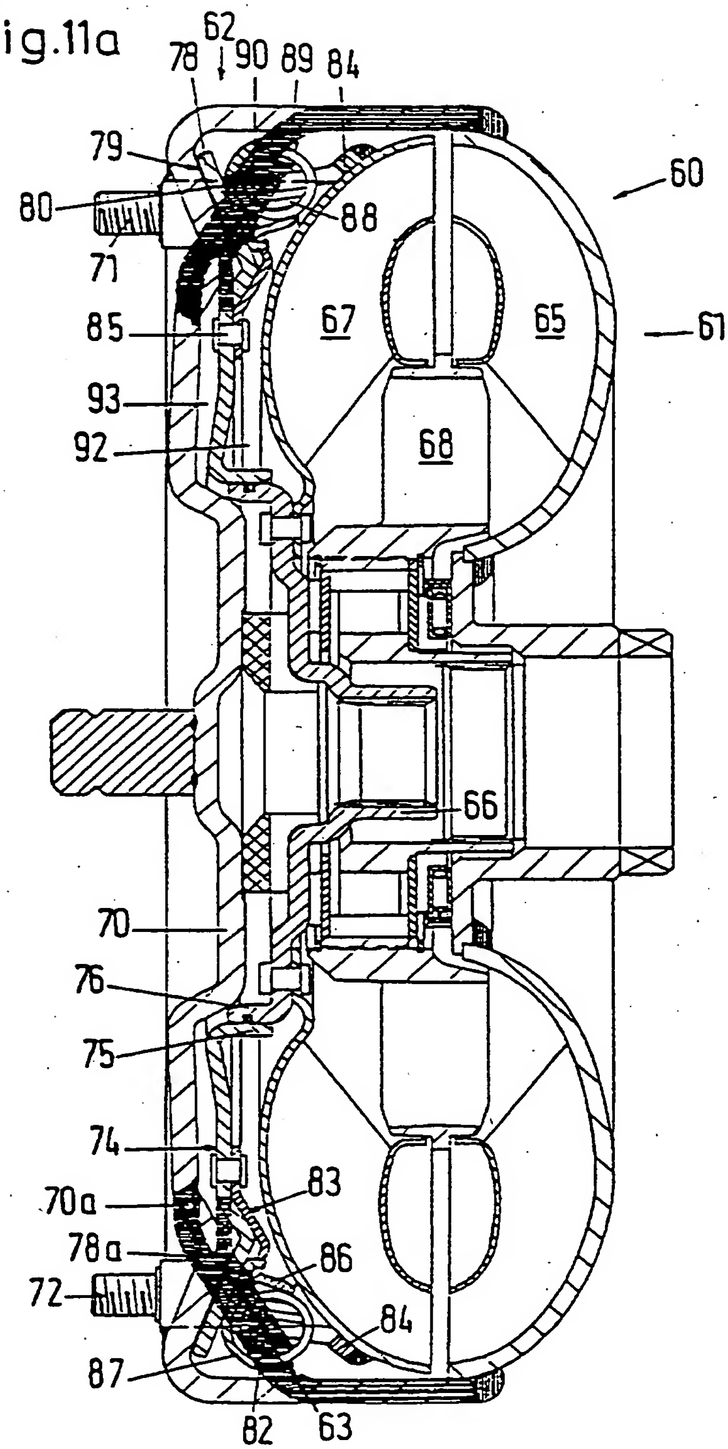

62
78
90
89
84
79
80
71
88
60
67
65
61
85
93
92
68
66
70
76
75
74
83
70a
78a
86
72
84
87
82
63

Fig.11b

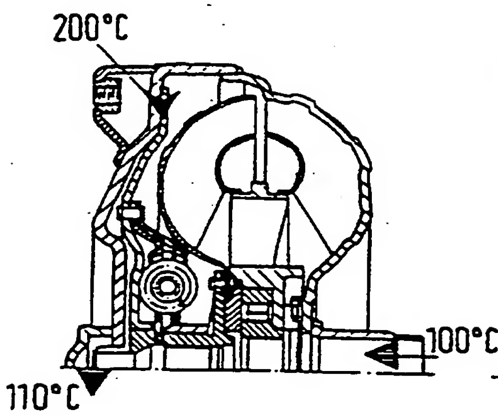


Fig.11c

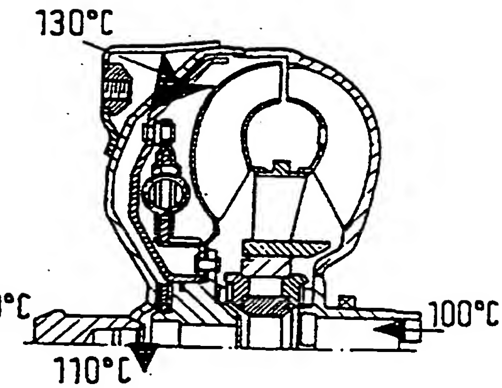


Fig.11d

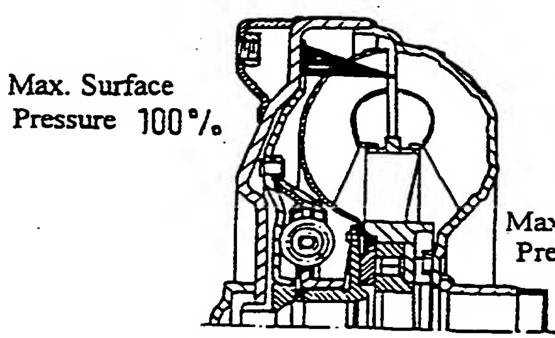


Fig.11e

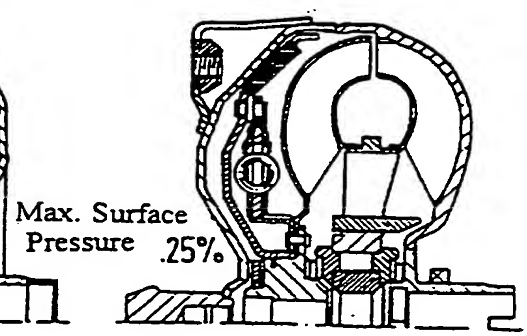


Fig.12

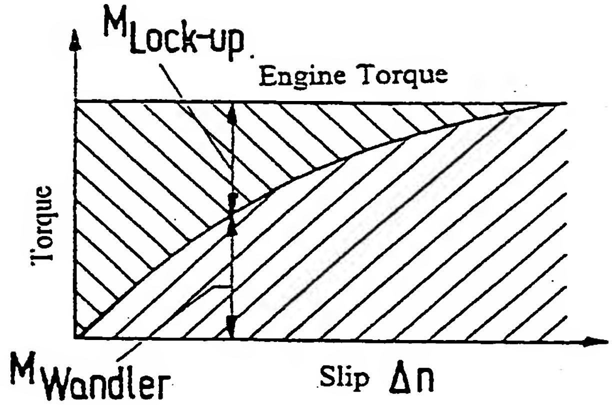

M Lock-up
Engine Torque
Torque
M Wandler
Slip Δn

Fig.13

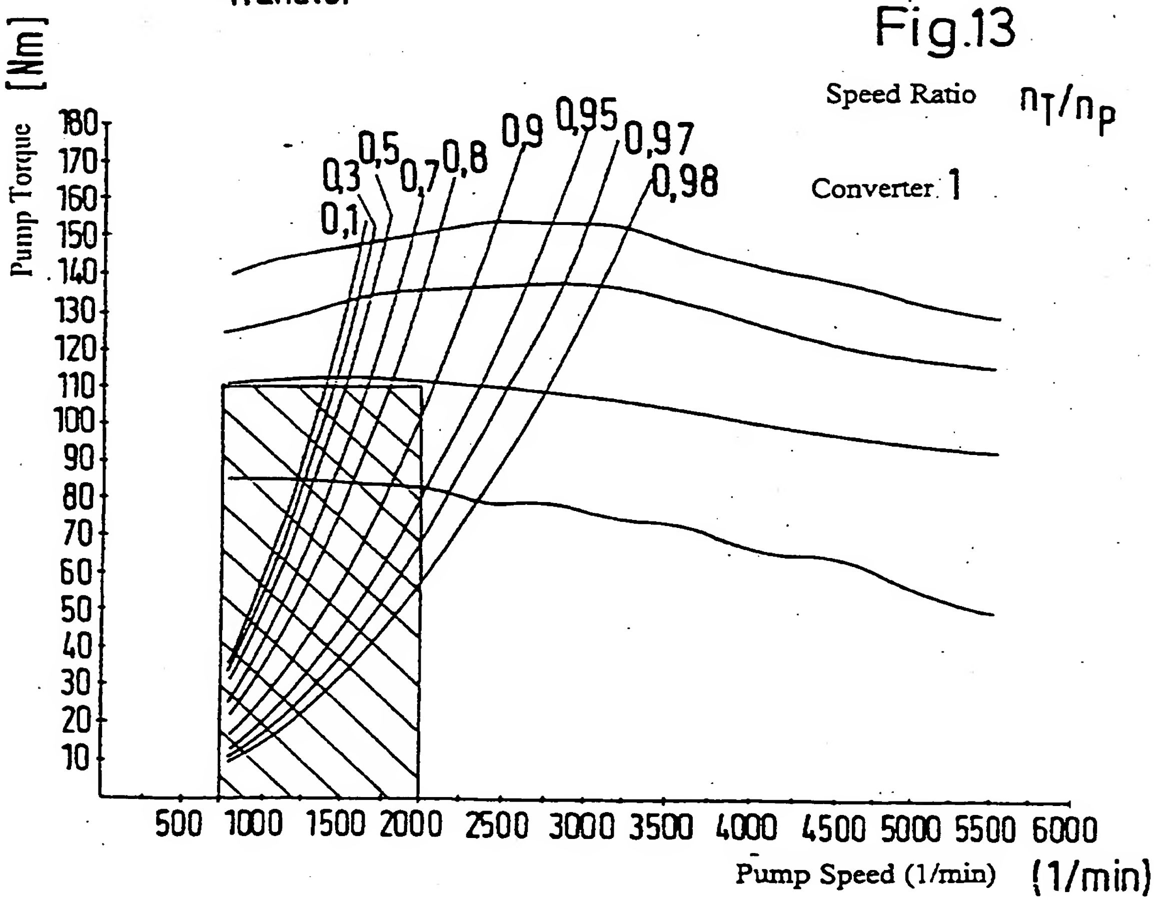

Speed Ratio n_T/n_P
Converter 1
0,1
0,3
0,5
0,7
0,8
0,9
0,95
0,97
0,98
Pump Torque [Nm]
180
170
160
150
140
130
120
110
100
90
80
70
60
50
40
30
20
10
500 1000 1500 2000 2500 3000 3500 4000 4500 5000 5500 6000
Pump Speed (1/min) (1/min)

Fig.14

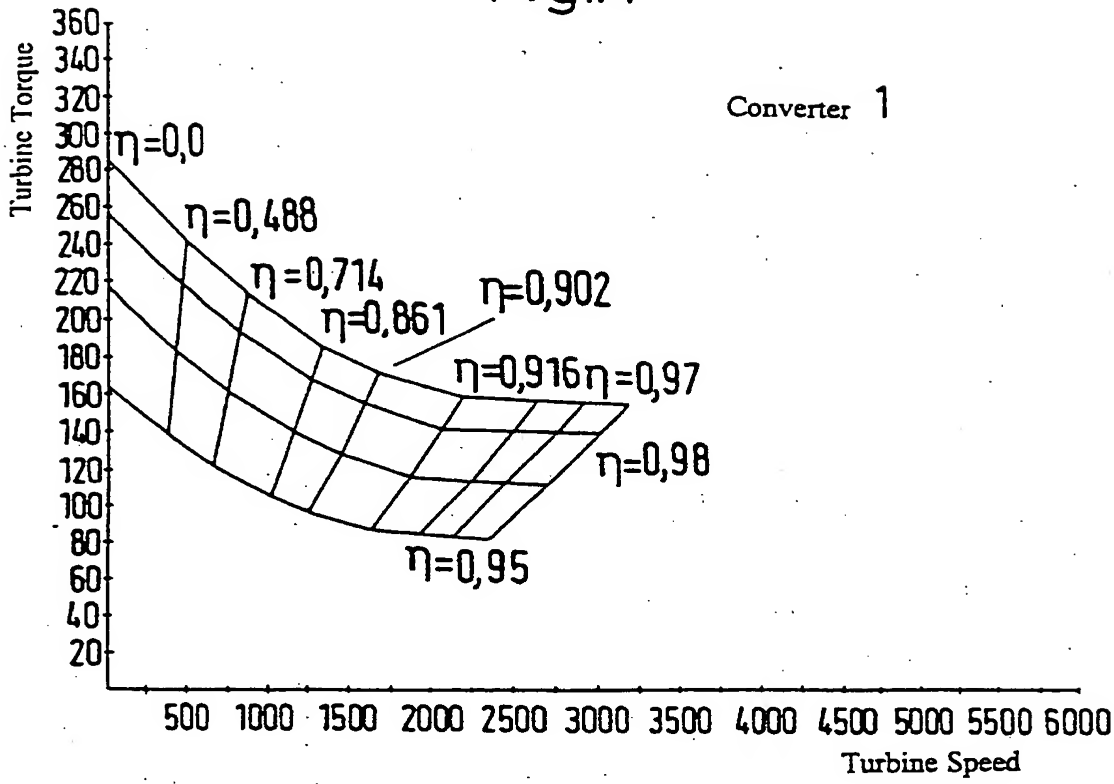

Turbine Torque
360
340
320
300
280
260
240
220
200
180
160
140
120
100
80
60
40
20
Converter 1
η=0,0
η=0,488
η=0,714
η=0,861
η=0,902
η=0,916
η=0,97
η=0,98
η=0,95
500 1000 1500 2000 2500 3000 3500 4000 4500 5000 5500 6000
Turbine Speed

Fig.15

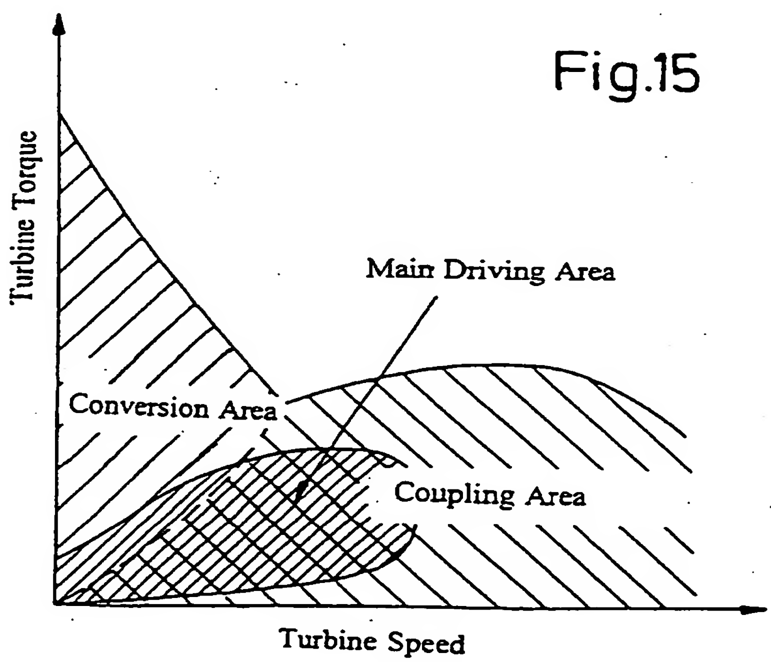

Turbine Torque
Main Driving Area
Conversion Area
Coupling Area
Turbine Speed

Fig.16

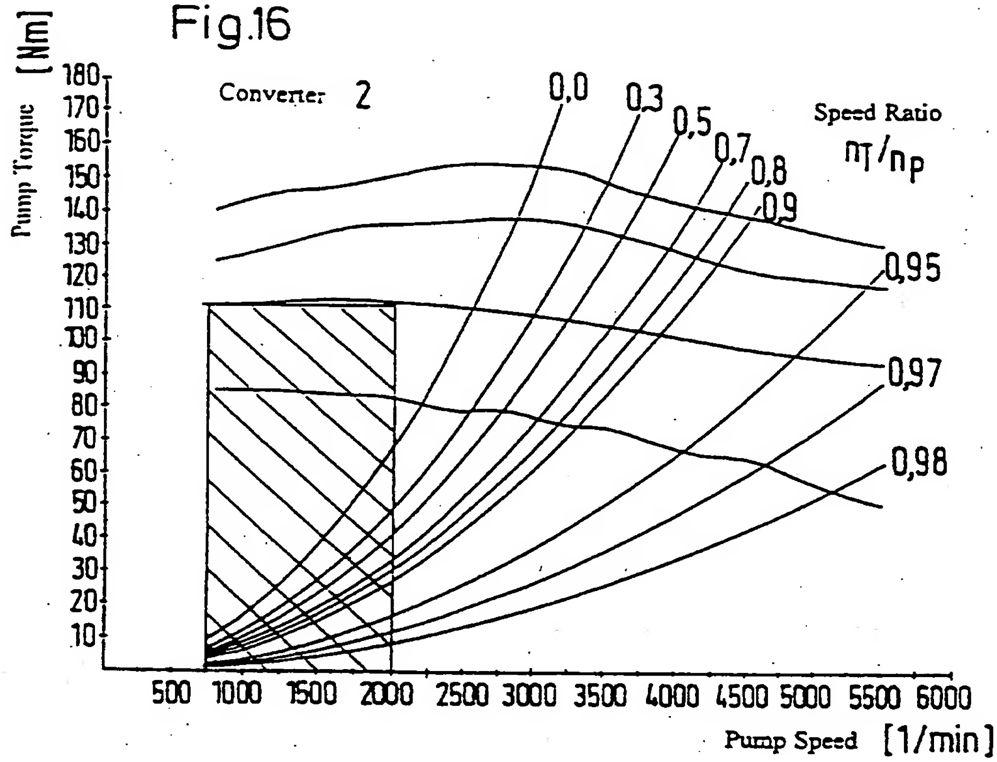

Pump Torque [Nm]
180
170
160
150
140
130
120
110
100
90
80
70
60
50
40
30
20
10
Converter 2
0,0
0,3
0,5
0,7
0,8
0,9
Speed Ratio
n_T/n_P
0,95
0,97
0,98
500 1000 1500 2000 2500 3000 3500 4000 4500 5000 5500 6000
Pump Speed [1/min]

Fig.17

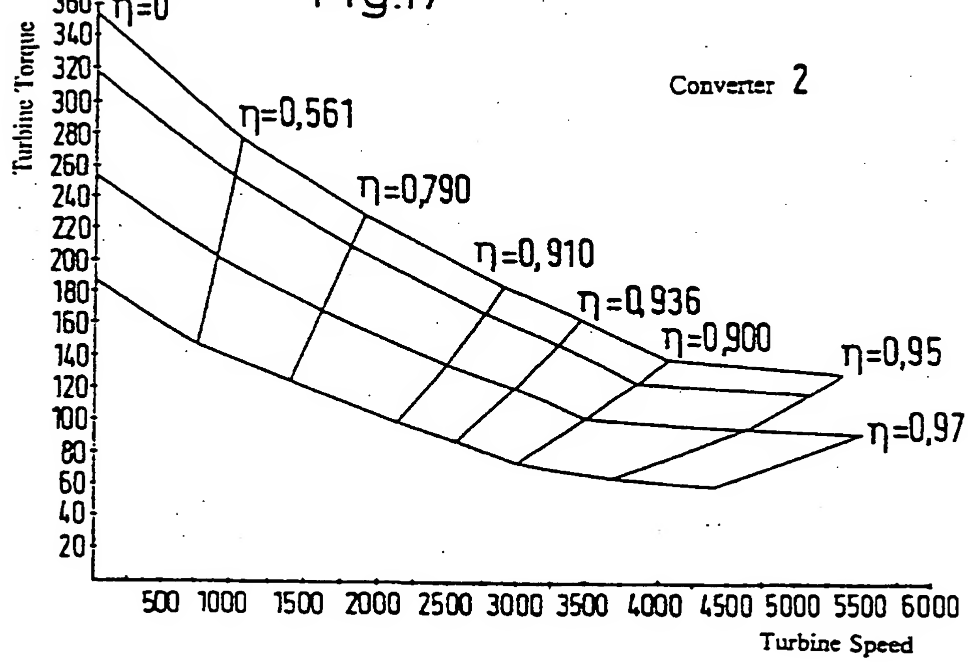

Turbine Torque
360
340
320
300
280
260
240
220
200
180
160
140
120
100
80
60
40
20
η=0
η=0,561
η=0,790
η=0,910
η=0,936
η=0,900
η=0,95
η=0,97
Converter 2
500 1000 1500 2000 2500 3000 3500 4000 4500 5000 5500 6000
Turbine Speed

Fig. 18

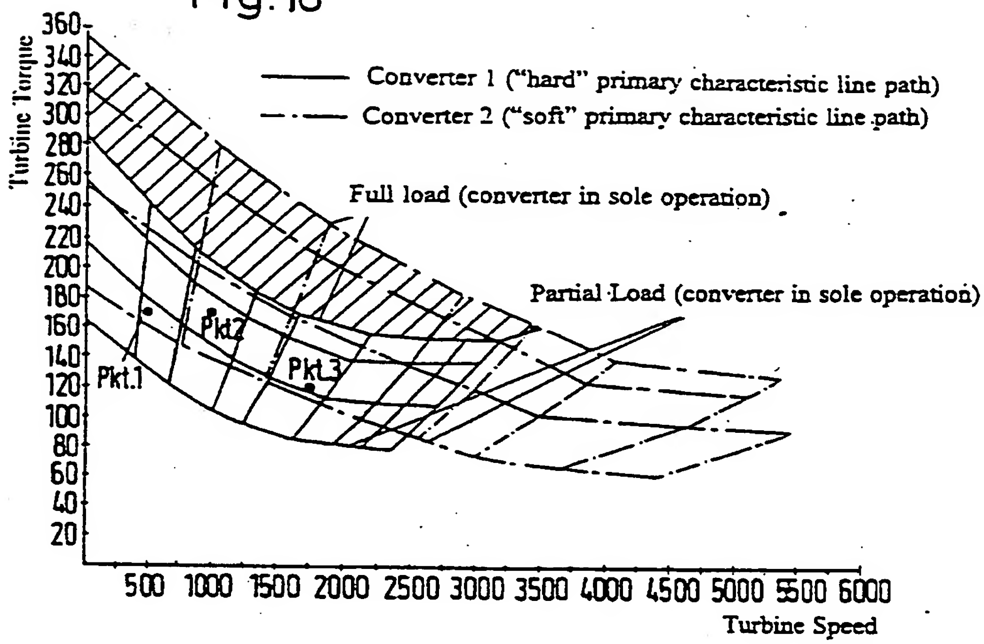

Turbine Torque
360
340
320
300
280
260
240
220
200
180
160
140
120
100
80
60
40
20
Converter 1 ("hard" primary characteristic line path)
Converter 2 ("soft" primary characteristic line path)
Full load (converter in sole operation)
Partial Load (converter in sole operation)
Pkt.1
Pkt.2
Pkt.3
500 1000 1500 2000 2500 3000 3500 4000 4500 5000 5500 6000
Turbine Speed

Fig. 19

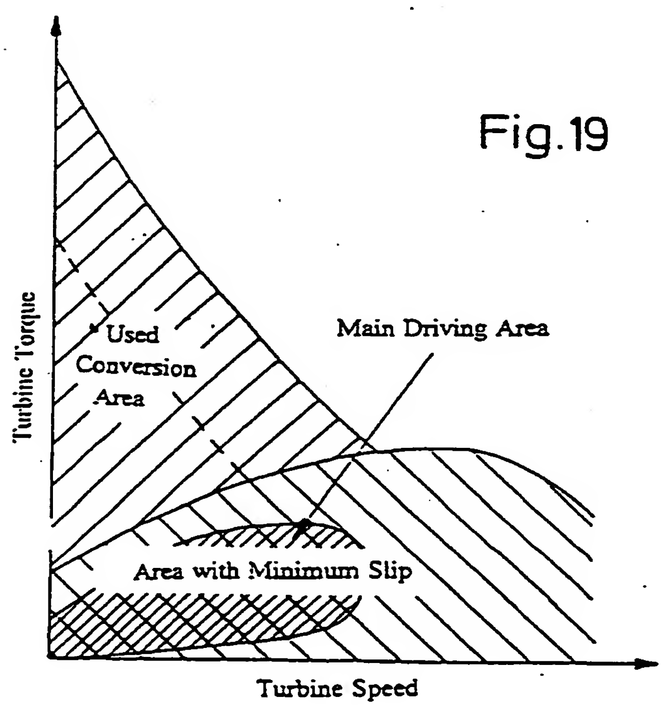

Turbine Torque
Used
Conversion
Area
Main Driving Area
Area with Minimum Slip
Turbine Speed

Fig.20

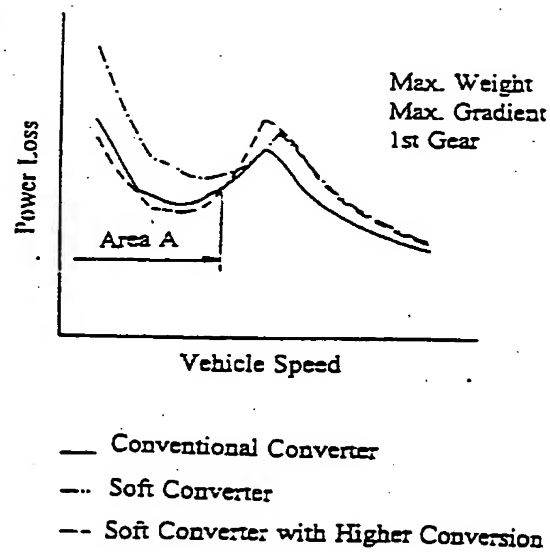


Fig.21

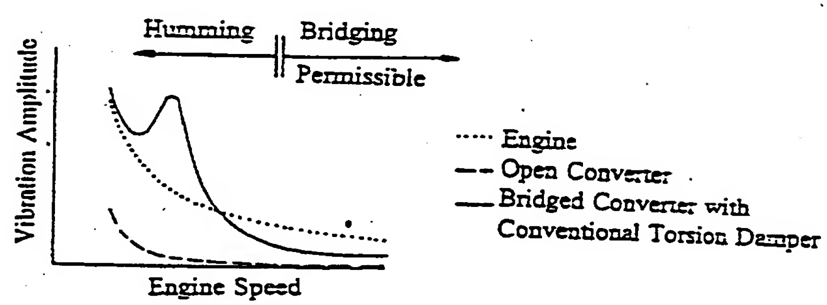


Fig.22

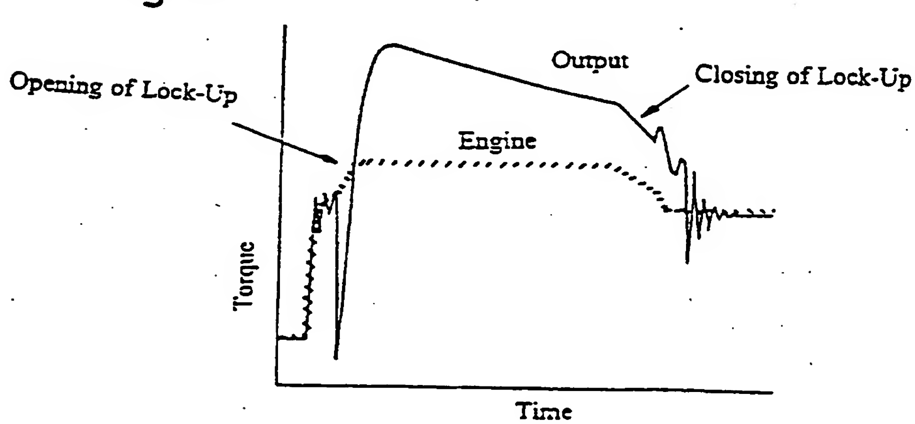


Fig.23

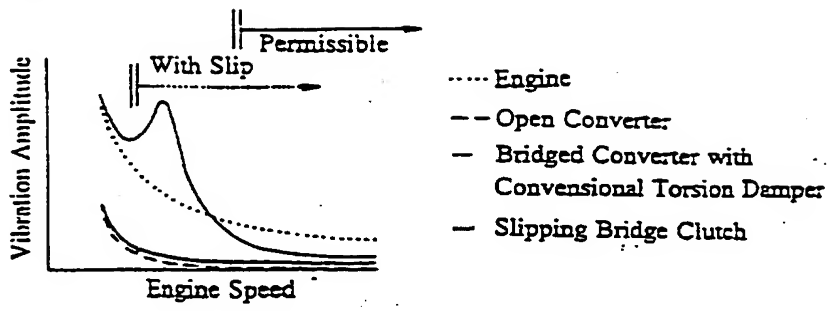


Fig.24

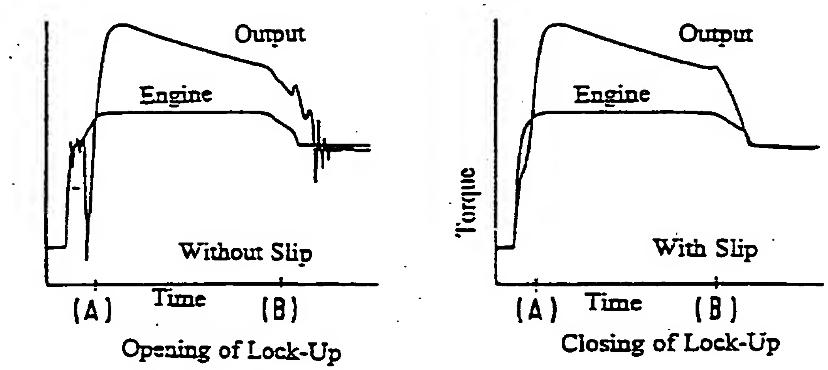
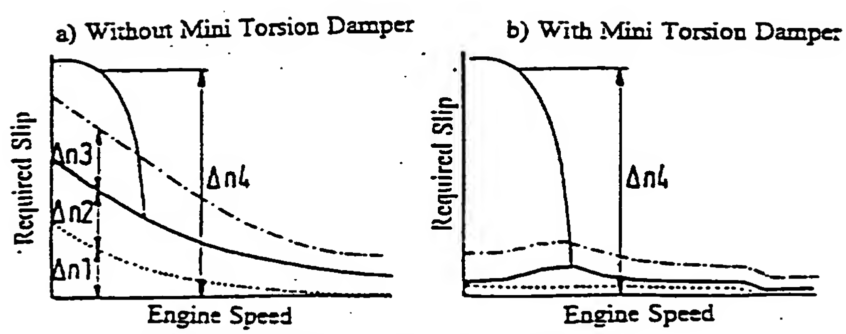


Fig.25



···· Δn1: Minimum Slip to Prevent Humming

— Δn2: Slip which has been set to Prevent Understepping Minimum Slip with Non-Stationary Processes

—·- Δn3: Slip which can be Produced with Non-Stationary Processes

— Δn4: Slip which is Necessary t Avoid Pressing Engine to Strongly

Fig.26

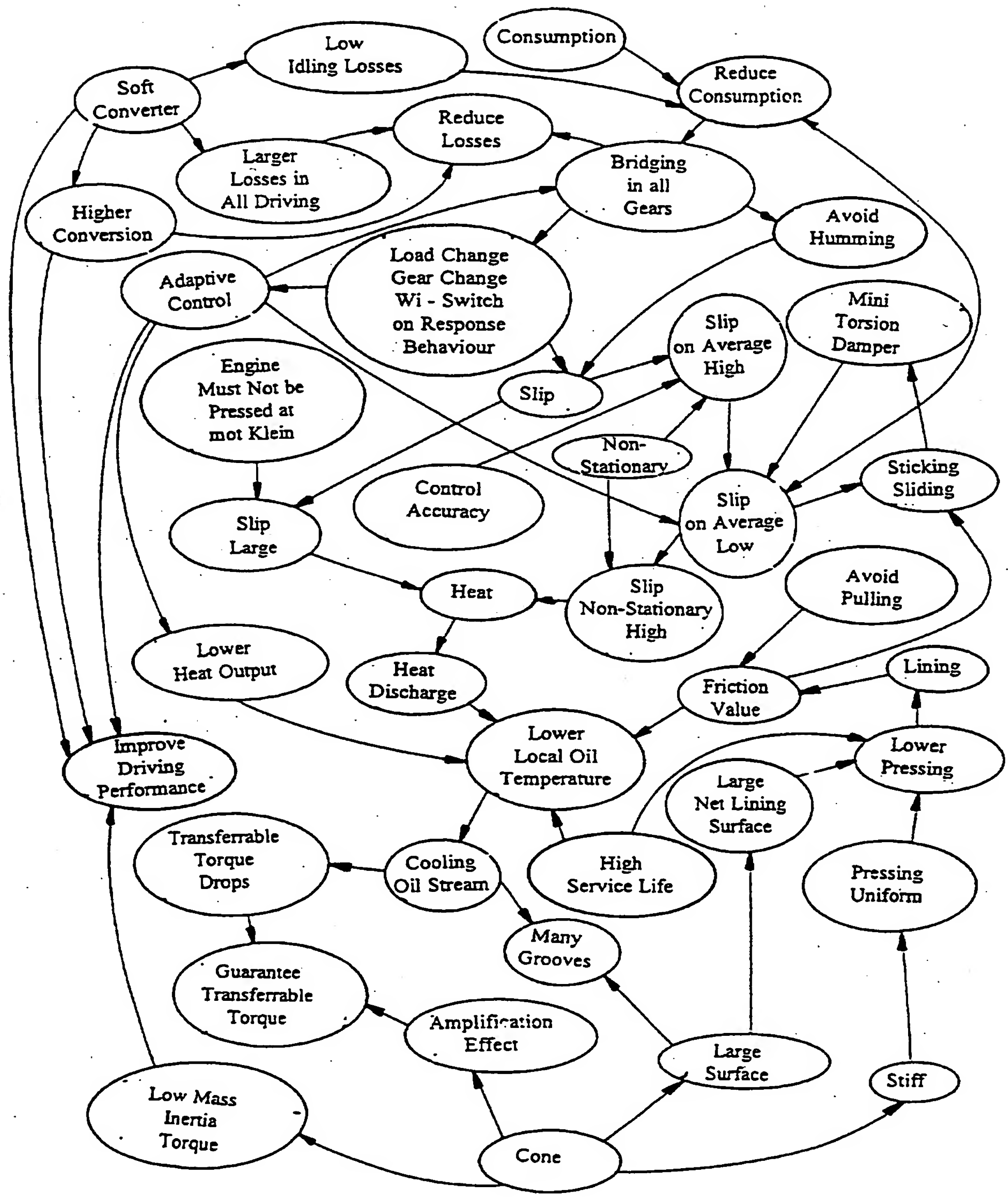

Consumption
Low Idling Losses
Reduce Consumption
Soft Converter
Reduce Losses
Larger Losses in All Driving
Bridging in all Gears
Higher Conversion
Avoid Humming
Adaptive Control
Load Change Gear Change Wi - Switch on Response Behaviour
Mini Torsion Damper
Slip on Average High
Engine Must Not be Pressed at mot Klein
Slip
Non-Stationary
Sticking Sliding
Control Accuracy
Slip on Average Low
Slip Large
Heat
Slip Non-Stationary High
Avoid Pulling
Lower Heat Output
Heat Discharge
Lining
Friction Value
Lower Local Oil Temperature
Lower Pressing
Improve Driving Performance
Large Net Lining Surface
Transferrable Torque Drops
Cooling Oil Stream
High Service Life
Pressing Uniform
Many Grooves
Guarantee Transferrable Torque
Amplification Effect
Large Surface
Stiff
Low Mass Inertia Torque
Cone

Fig.27

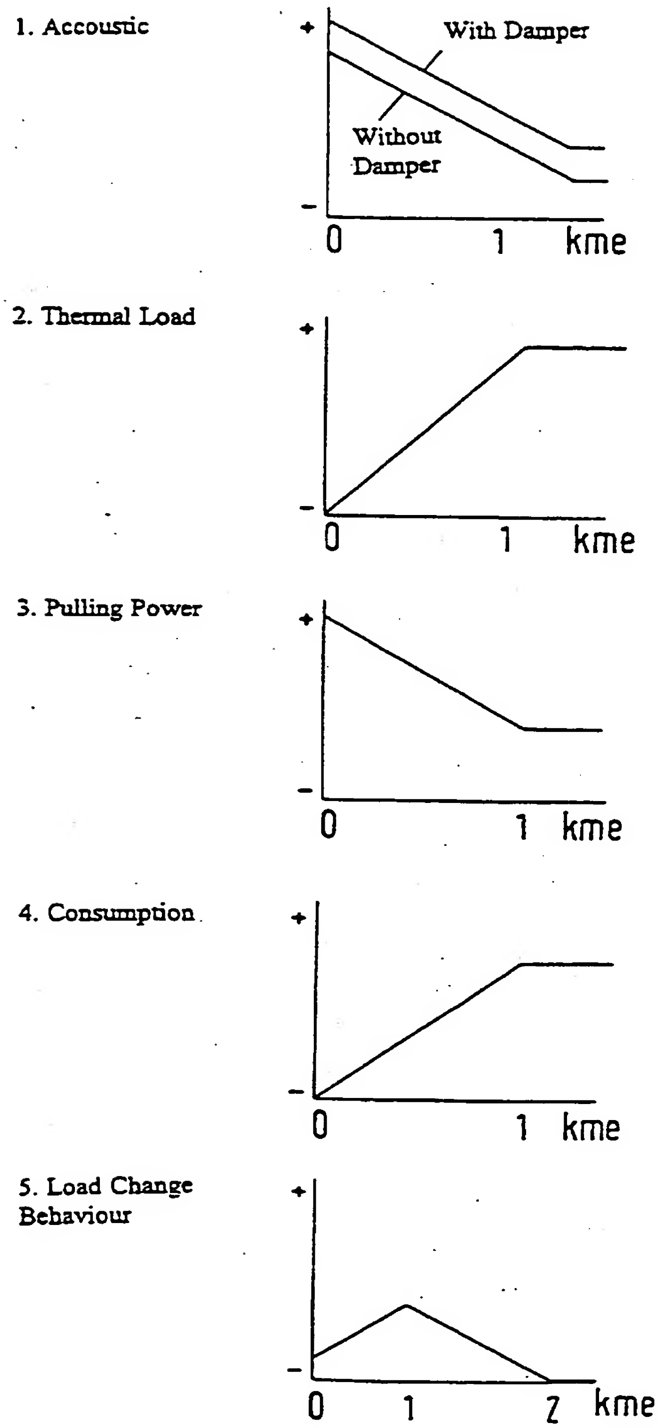

1. Accoustic
+
With Damper
Without Damper
-
0
1
kme
2. Thermal Load
+
-
0
1
kme
3. Pulling Power
+
-
0
1
kme
4. Consumption
+
-
0
1
kme
5. Load Change Behaviour
+
-
0
1
2
kme

Fig.28

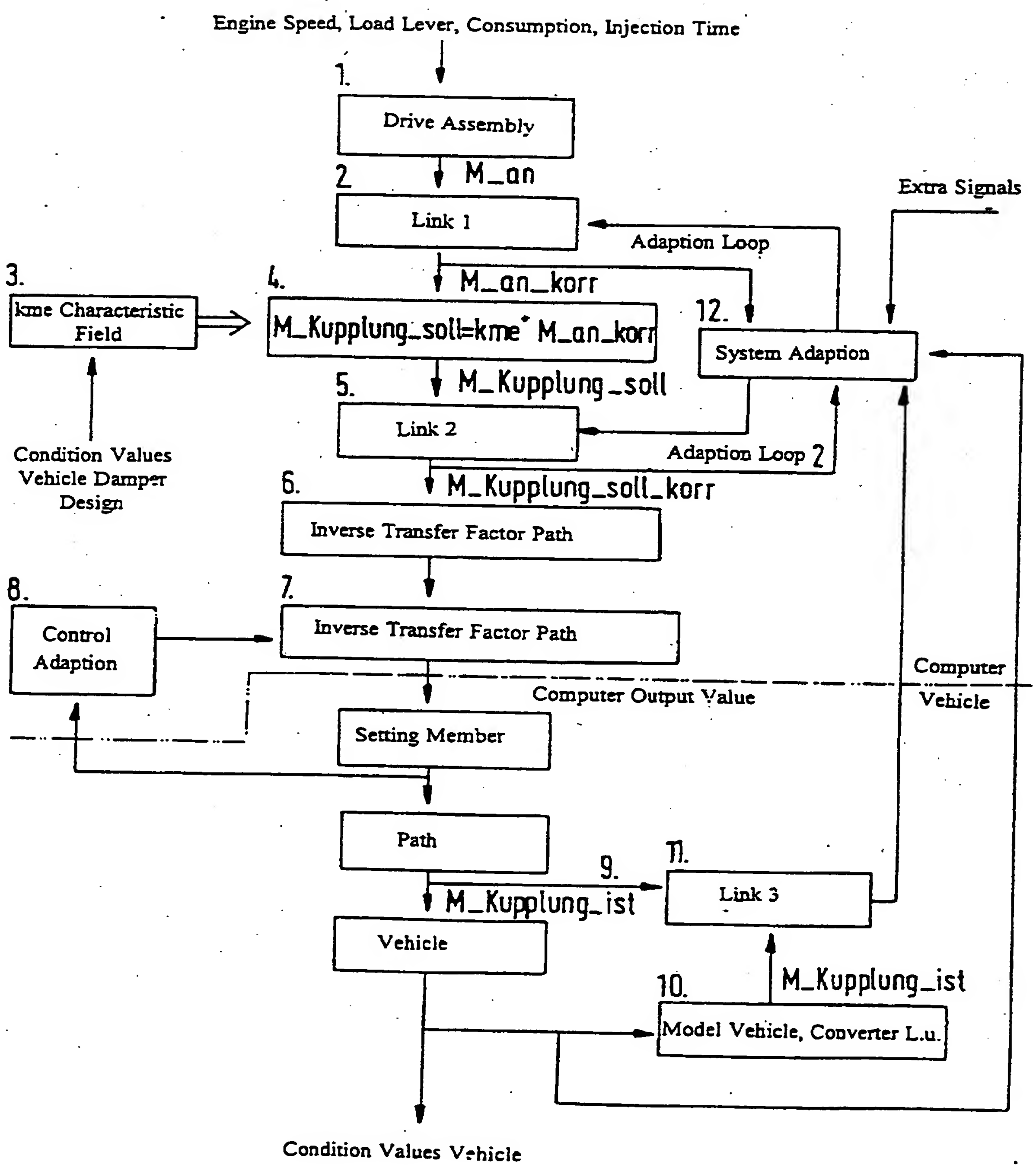

Engine Speed, Load Lever, Consumption, Injection Time
1.
Drive Assembly
2
M_an
Link 1
Adaption Loop
Extra Signals
3.
kme Characteristic Field
4.
M_an_korr
M_Kupplung_soll=kme* M_an_korr
12.
System Adaption
5.
M_Kupplung_soll
Link 2
Adaption Loop 2
Condition Values Vehicle Damper Design
6.
M_Kupplung_soll_korr
Inverse Transfer Factor Path
8.
7.
Control Adaption
Inverse Transfer Factor Path
Computer
Computer Output Value
Vehicle
Setting Member
Path
9.
11.
Link 3
M_Kupplung_ist
Vehicle
10.
M_Kupplung_ist
Model Vehicle, Converter L.u.
Condition Values Vehicle

Fig.29

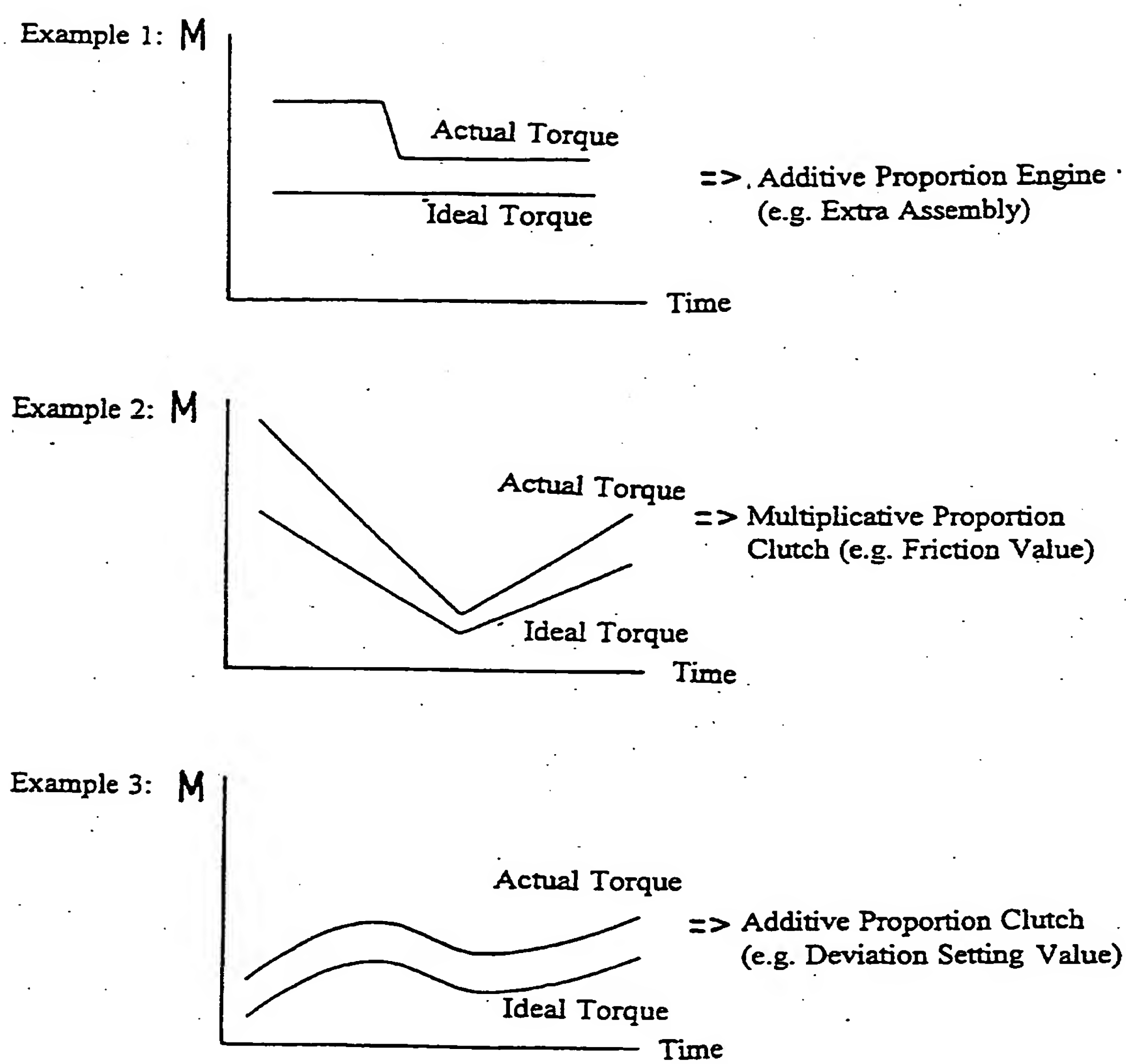

Example 1: M
Actual Torque
Ideal Torque
Time
=> Additive Proportion Engine
(e.g. Extra Assembly)
Example 2: M
Actual Torque
Ideal Torque
Time
=> Multiplicative Proportion
Clutch (e.g. Friction Value)
Example 3: M
Actual Torque
Ideal Torque
Time
=> Additive Proportion Clutch
(e.g. Deviation Setting Value)

Fig.30

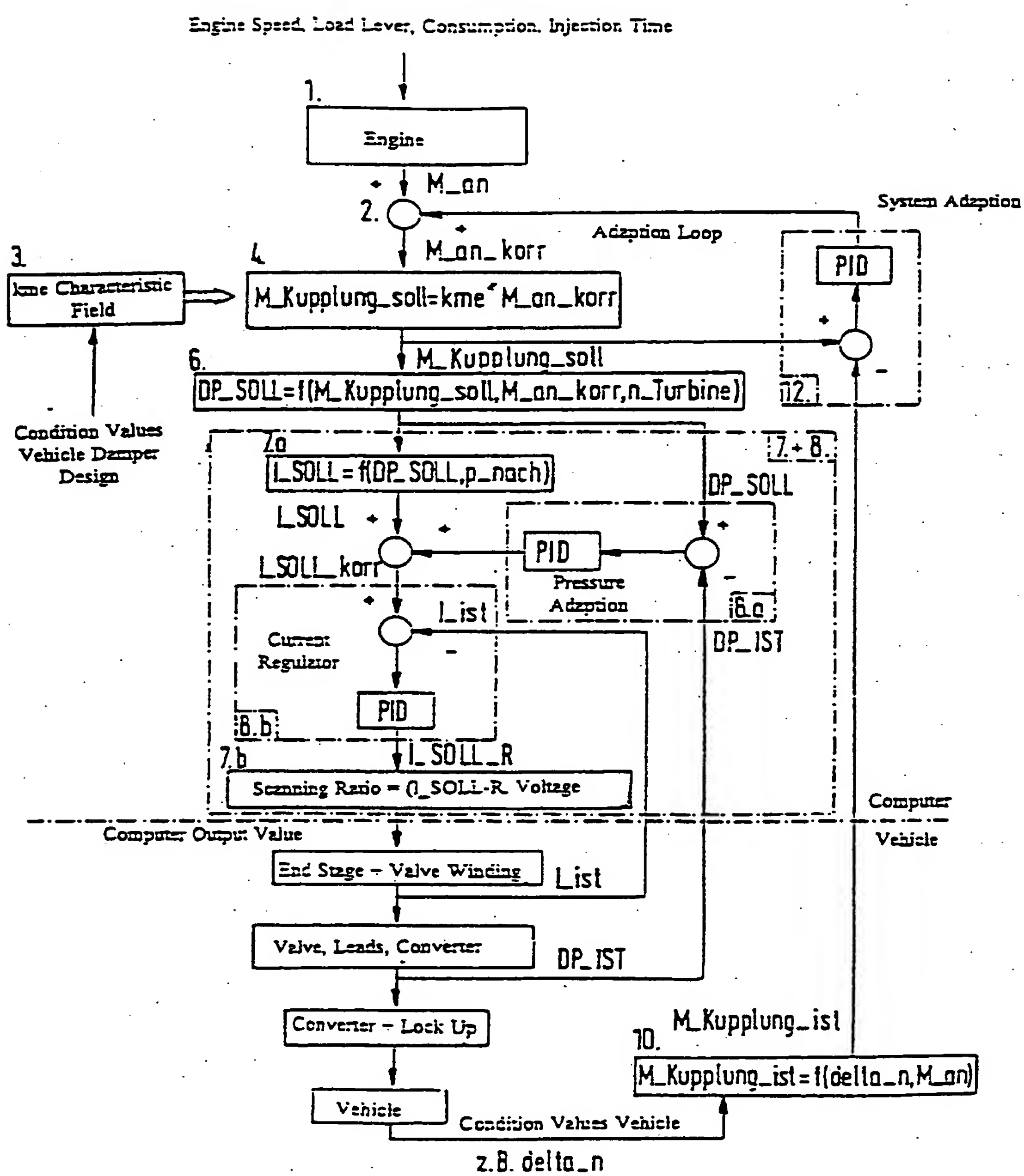

Engine Speed, Load Lever, Consumption, Injection Time
1.
Engine
M_an
2.
M_an_korr
Adaption Loop
System Adaption
PID
3.
kme Characteristic Field
4.
M_Kupplung_soll=kme*M_an_korr
M_Kupplung_soll
6.
DP_SOLL=f(M_Kupplung_soll,M_an_korr,n_Turbine)
12.
Condition Values Vehicle Damper Design
7a
I_SOLL=f(DP_SOLL,p_nach)
7.+8.
DP_SOLL
I_SOLL
PID
I_SOLL_korr
Pressure Adaption
8.a
I_ist
Current Regulator
DP_IST
PID
8.b
7.b
I_SOLL_R
Scanning Ratio = (I_SOLL-R, Voltage
Computer
Computer Output Value
Vehicle
End Stage + Valve Winding
I_ist
Valve, Leads, Converter
DP_IST
Converter + Lock Up
M_Kupplung_ist
10.
M_Kupplung_ist=f(delta_n,M_an)
Vehicle
Condition Values Vehicle
z.B. delta_n

Fig.31

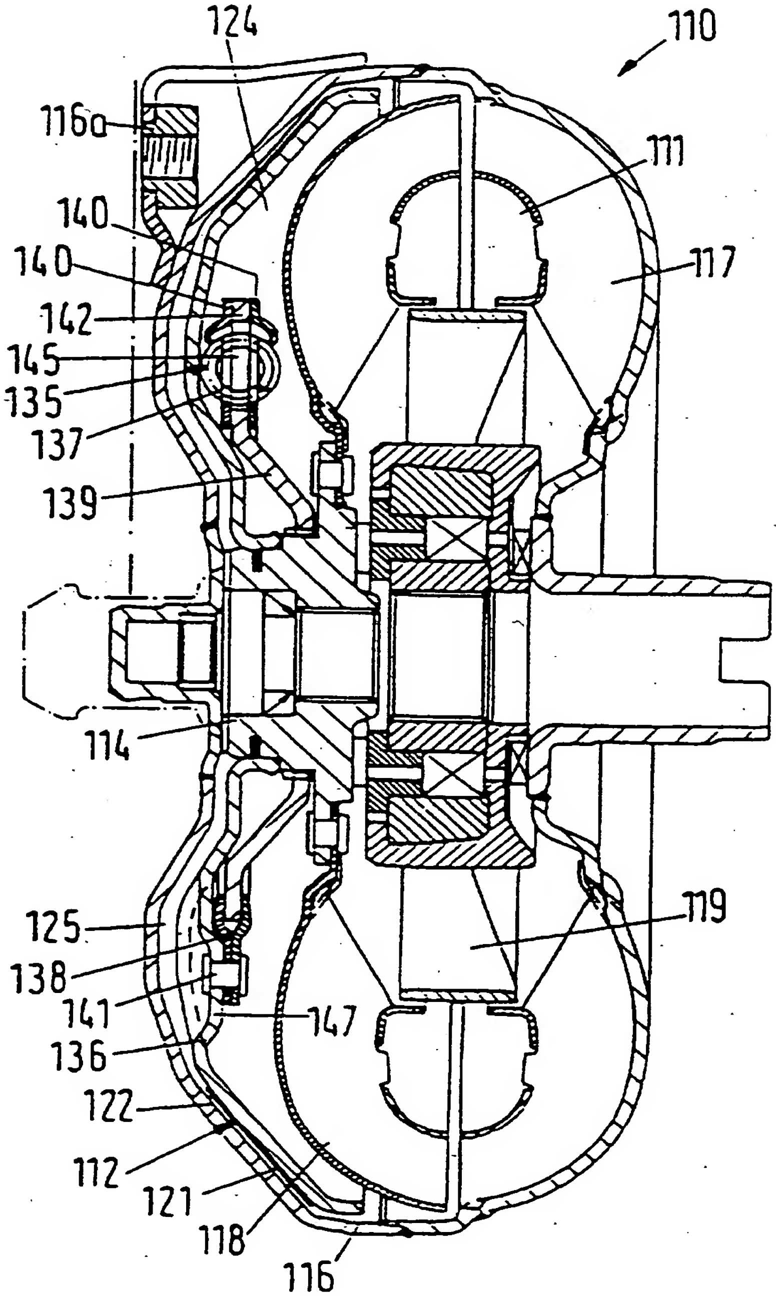

124
110
116a
111
140
117
140
142
145
135
137
139
114
119
125
138
141
147
136
122
112
121
118
116

Fig.32

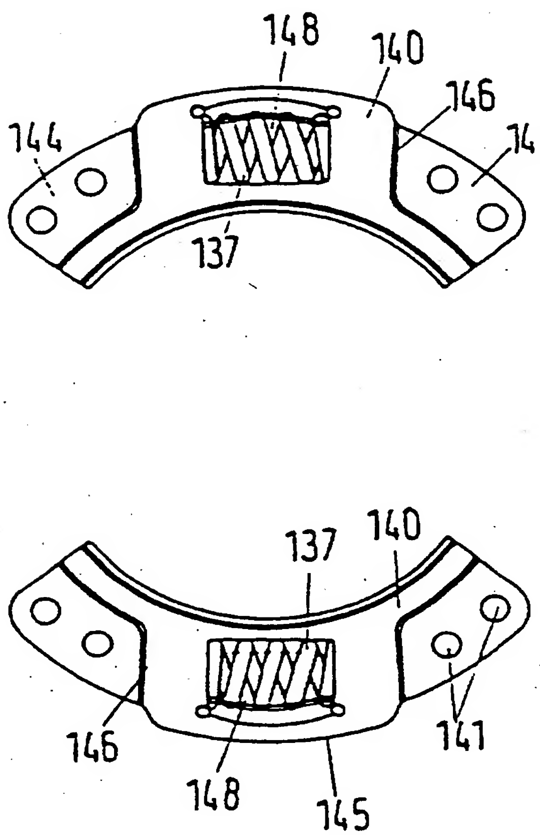

148
140
146
144
144
137
137
140
146
148
145
141

Fig.33

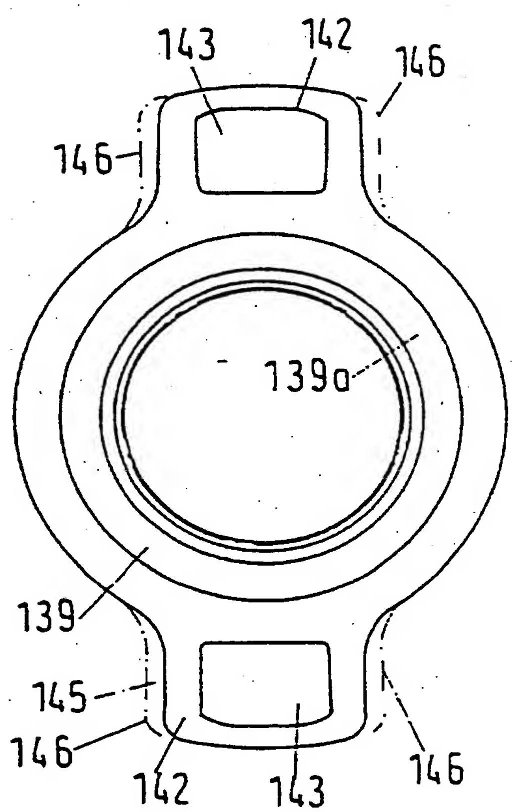

143
142
146
146
139a
139
145
146
142
143
146

Fig.35

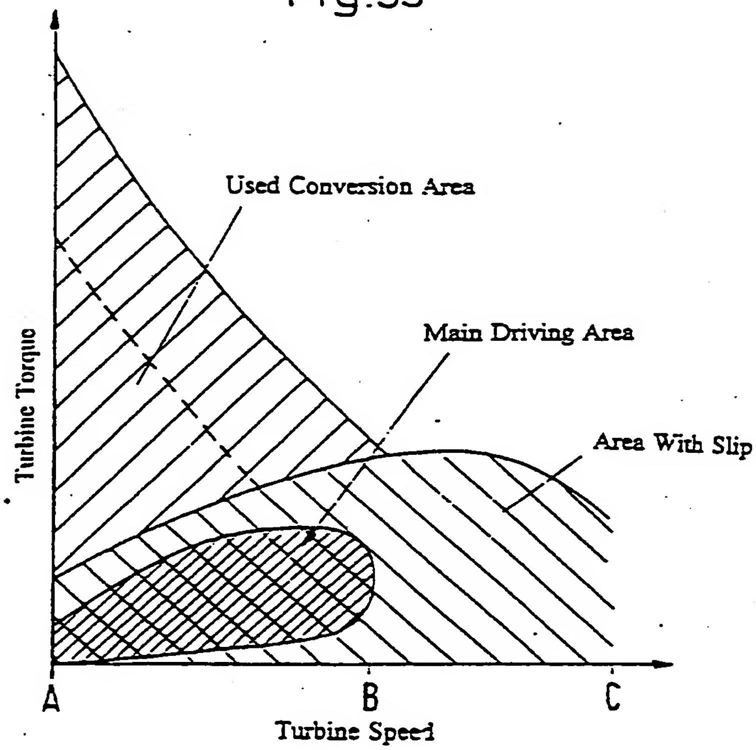

Used Conversion Area
Main Driving Area
Area With Slip
Turbine Torque
A
B
C
Turbine Speed

Fig.34

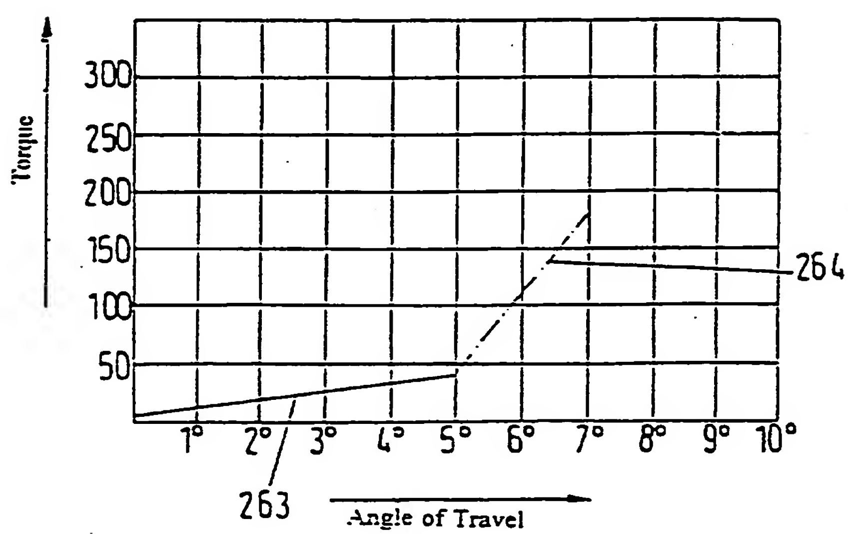

Torque
300
250
200
150
100
50
1°
2°
3°
4°
5°
6°
7°
8°
9°
10°
263
264
Angle of Travel

Fig.36

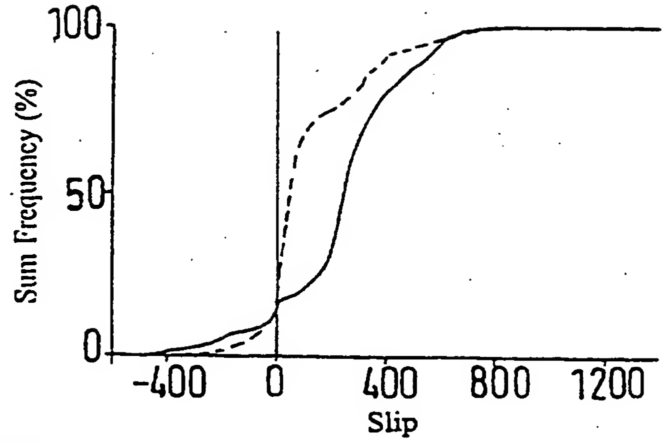


Fig.37

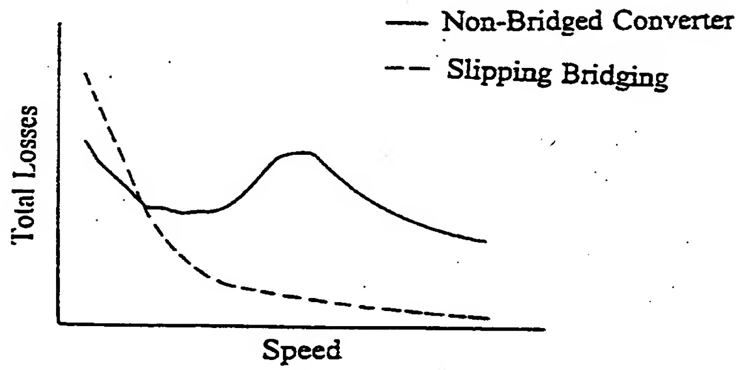


Fig.38

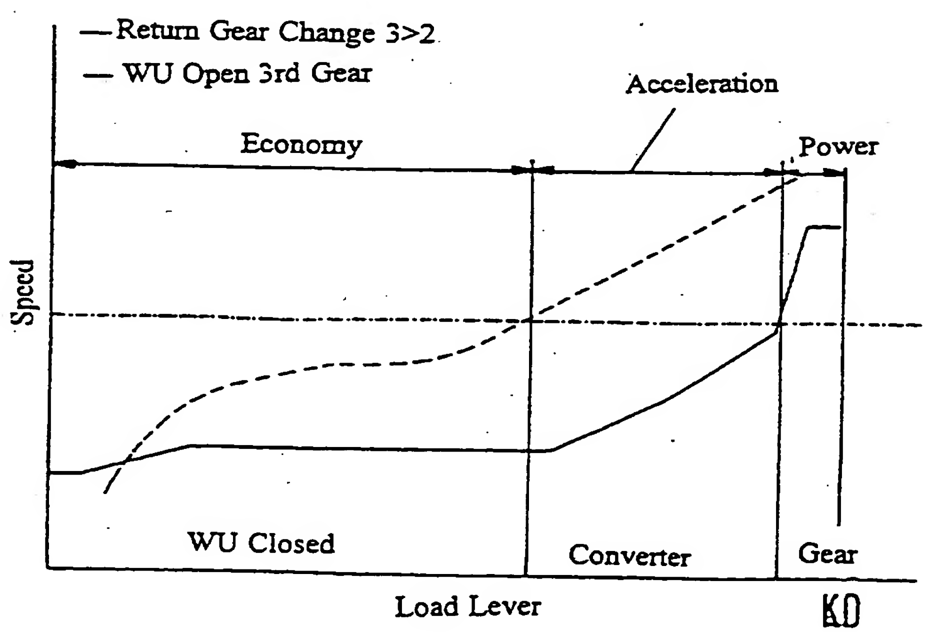
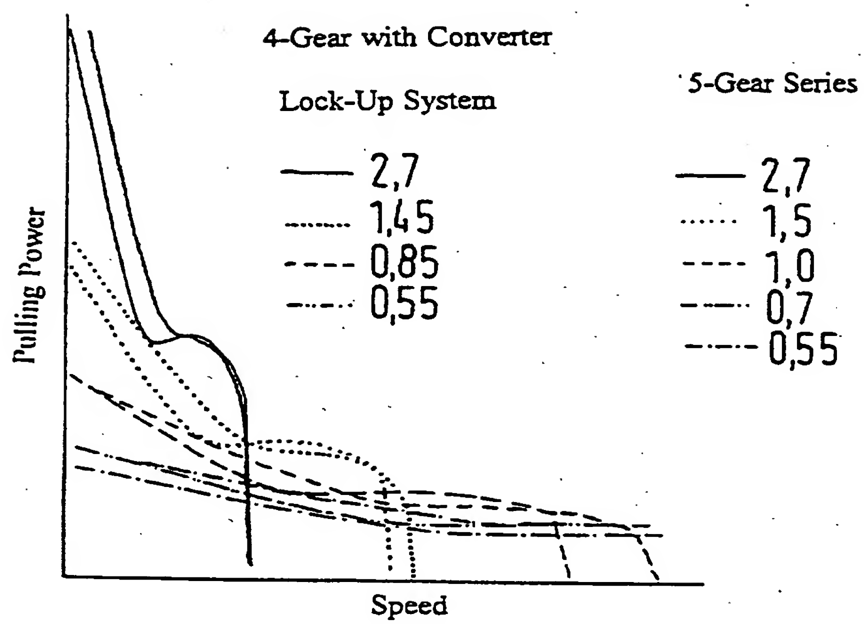


Fig.39



4-Gear with Converter
Lock-Up System
2,7
1,45
0,85
0,55
5-Gear Series
2,7
1,5
1,0
0,7
0,55
Pulling Power
Speed

2320531

Lock-up Clutch for a Torque Converter

The invention relates to a lock-up clutch for the torque converter of a torque transfer system.

Methods or processes for controlling torque transfer systems wherein the torque to be transferred by the friction clutch is set by deliberately adjusting the differential pressure between the pressure chambers of a friction clutch mounted in parallel with the converter and bridging same, are known.

Thus for example in DE-OS 31 30 871 in connection with a torque transfer system of the kind mentioned above a regulating method is described where the slip values occurring between the drive and output are measured, compared with the preset ideal slip values and any differences detected are counter-regulated. The latter is brought about by changing the difference between the flow medium pressurised biasings of the two pressure chambers of a friction clutch. This is thus a regulating method based on the standard slip regulation.

From US-PS 5 029 087 a regulating method is also already known for converters with parallel mounted friction clutch wherein the slip on the clutch is measured, compared with predetermined ideal slip values and the differential pressure between the two pressure chambers of the friction clutch is changed in dependence on the fixed deviations of the differential pressure. This is also a typical slip regulation wherein measured deviations from the predetermined slip values are counter-regulated.

From US PS 4 577 737 a method is known for influencing a torque transfer system of the kind mentioned above wherein the torque transfer through a hydrodynamic converter is

measured direct by means of a torque sensor and the torque transfer is fixed in dependence on the operating state of the drive machine. The end of the friction clutch bridging the converter is thereby regulated so that the required torque transfer is guaranteed.

With this control method the torque transferred by the converter, like the adjusting slip, can naturally only then be measured and influenced after it has been set. This here is also a regulation concept related to the slip regulation, although here the torque is used which is to be transferred by the converter.

Systems of this kind for deliberately influencing the torque transferred by the friction clutch of a torque transfer system of the kind discussed above have not proved satisfactory or at least not completely satisfactory in practice.

Thus in the case of a slip regulation according to the system it is only possible to react to changes in the slip when they are measured, thus when they already exist. This has disadvantages particularly in the case of dynamic processes.

Thus a cancellation of the torque on the part of the drive machine causes a reduction in the slip in the torque transfer system. In order to avoid a state where the friction clutch sticks and thus of unrestricted transfer of the torque fluctuations from the drive machine to the remaining drive train a cancellation of the torque transferred by the friction clutch must take place. The dynamics of a regulation is however in practice restricted by delay and dead times conditioned by the system so that a minimum slip is necessary which from experience cannot be

below 50 rpm.

Furthermore driving situations do exist where a time-optimized control design is a hindrance.

Conditioned by the distribution of the rotary masses in the vehicle the speed at the input of the gear-change system or infinite gearing and thus at the output of the torque transfer system when changing up gear or changing translation is lowered whilst the speed at the output of the gearing remains relatively constant. The reduction in the output speed of the torque transfer system is connected with an increase in the slip whereby again, conditioned by the behaviour of the hydrodynamic converter, an increased torque is required at the input of the torque transfer system. However the drive assembly is not ready for this increased torque at this time. Consequently the drive assembly is braked and a slip is independently set again to a lower level when the biasing of the friction clutch when changing up gear or changing translation is kept constant. A time-optimized regulator will however endeavour to counteract the increase in slip by increasing the force biasing of the friction clutch which at the end of the gear change leads to a sticking of the friction clutch and thus to a transfer of the torque irregularity of the drive assembly to the remaining drive train.

Finally from DE-PS 37 12 223 a control method for a torque transfer system of the kind already mentioned is known wherein in a predetermined driving speed range the clutch engagement force is controlled in dependence on the throttle valve opening degree so that a slip can be set between the drive and output. As opposed to the slip control discussed above this here is a control which looking ahead sets in dependence on the throttle valve opening degree a force

biasing of the friction clutch at which the slip between the drive and output of the torque transfer system is adjusted in dependence on this force biasing.

However this control is unsatisfactory since the torque transferred by the friction clutch is not solely dependent on the clutch engagement force but also on the friction value of the friction lining which in turn as is known is subject to severe fluctuations in dependence on the temperature, slip speed, behaviour of the oil used and other factors. This means that even with this control variation a minimum slip speed must be observed in order to ensure even with fluctuations in the system behaviour a slip speed which is large enough to isolate vibrations. All the systems known up until now have the drawback that it is only possible to work with relatively large minimum slip speeds of more than 50 rpm. This on the one hand brings hardly any advantages in fuel consumption compared to the non-bridged converter and on the other hand makes it difficult for the power losses occurring at the friction clutch to be curbed.

Consequently the object of the invention is inter alia to provide an improved method for controlling a torque transfer system which allows the slip speeds to be set clearly lower than 50 rpm in all driving situations of a vehicle with converter and output gearbox connected at the output side.

Friction clutches for bridging the hydrodynamic torque converter of such torque transfer systems are indeed generally known.

In the case of lock-up clutches with flat friction faces the friction radius is dependent on the pressure biasing and in view of the lower stiffness does not guarantee a uniform

pressure distribution over the entire friction lining. This leads in the case of slipping clutches to the partial overheating of the friction lining and thus to the destruction thereof as well as of the oil located in this area (automatic transmission fluid = ATF).

Furthermore the torque transferable by the friction clutch is directly dependent on the friction radius which in connection with the oil pressure available in the automatic gear box requires a minimum radial structural space.

A converter lock-up clutch of the kind described above however, requires a larger axial structural space which in many vehicle gearboxes is just not available particularly if with the piston damper unit elastic damping means are to be mounted on a large radius. These mechanical damping means are necessary in order to guarantee even in areas of very large vibration activity on the part of the drive machine an optimum vibration isolation even with low slip speeds.

Starting from this prior art a further object of the invention is to provide an improved lock-up clutch of the kind and for the purpose mentioned.

According to the invention, there is provided a lock-up clutch for a hydrodynamic torque converter of a torque transfer system, with a pump wheel, turbine wheel, guide wheel and a converter cover which is central with the axis of rotation, connected rotationally secured with the pump wheel and encloses the turbine wheel, wherein a central ring piston mounted between the converter cover and turbine wheel is designed radially outwards as a conical clutch friction disc and has radially inwards a sealing hub which is set on a counter sealing hub which is connected rotationally secured with the turbine wheel.

The correct adjustment of the torque can be achieved by comparing the predetermined clutch torque with the clutch torque actually present at that time. The actual clutch torque can thereby be detected from the difference between the torque released by the engine and the converter torque. The converter torque can be detected from the input and output speeds (direct) of the converter. The converter output speed can however also be calculated from the gearbox output speed and the gear translation. To this end the converter characteristic field can be recorded in the computer.

The torque to be transferred by the clutch is fixed at least in a ratio to the engine torque wherein this ratio is determined at least by the k_{me} -factor which is determined in dependence on at least one of the operating conditions

permissible noise level (acoustics in the vehicle),
thermal load on the torque converter or lock-up clutch,
pulling power or acceleration of the vehicle,
load change behaviour of the vehicle or comfort,
consumption.

The aforesaid operating conditions or criteria which can be used to determine the k_{me}-factor are contradictory in part, ie they are opposed in relation to the k_{me}-factor so that when considering several of these criteria it is necessary to weigh up their priority. Thus for example the acoustics cannot be improved in any way by selecting a very small k_{me} factor because otherwise as a result of the high slip in the lock-up clutch an inadmissibly high thermal load may arise on the converter. There are thus border conditions which should not be overstepped or understepped. On the positive side of these border conditions however there remains a certain possibility of variation regarding the k_{me}-factor

wherein this k_{me}-factor can remain constant or however can be varied in dependence on the resulting operating conditions. This variation can take place step-wise or continuously between the boundary values. Advantageously the k_{me}-factor can be in the order of between 0.4 and 1.1 wherein it can be particularly expedient if this has a value between 0.7 and 0.95. At least in many operating areas of an internal combustion engine, preferably in the lower and where applicable also in the upper areas the k_{me}-factor can be selected so that the lock-up clutch can transfer the full net torque released by the internal combustion engine. With this type of procedure it is expedient if the lock-up clutch has a damper which is designed for the partial load area. A damper of this kind thus has a lock-up or stop torque which is less than the net torque released by the combustion engine. This stop torque can be in the order of between 30 and 60% of the net torque of the combustion engine. By using a vibration damper of this kind it is possible to at least partially counteract the acoustic problems which occur in the lower operating range of an internal combustion engine in connection with a comparatively large k_{me}-factor. As can be seen from the previous description the k_{me}-factor, which as already mentioned can be variable and can be calculated either in dependence on the situation parameters or operating parameters through a mathematical link or however can be recorded in a computer or processor in the form of a characteristic field or file, is the characteristic for the design of the vehicle.

The essential difference between the known slip controls in converter lock-up clutches and the torque control according to the invention lies in the fact that with the slip control the slip itself is the controlling value whilst with the torque control according to the invention the control or regulating value is a torque or a value representing this

torque, such as eg the biasing pressure or biasing force of the lock-up clutch. With the adaption according to the invention of the value or setting parameter representing the torque to be transferred this value or setting parameter is in practice corrected in the manner of a breakdown factor observation. This means the deviation of the torque actually transferred in relation to the torque to be transferred is calculated through a model and can be corrected accordingly. This correction can take place for example through a PID-portion or only an I-portion, eg by addition. It is however also possible to make a corresponding correction through a multiplicative factor or however a correction through both a multiplicative and additive portion or factor is also advantageous.

With the invention it is therefore a question of dividing the input torque into a hydraulic portion which is to be transferred by the converter and a mechanical portion which is to be transferred by the friction clutch. For the infinite control the lock-up clutch is biased with variable force which is controlled by an intelligent control so that an optimum division of converter torque and lock-up torque is produced for each driving situation.

It is characteristic for the control method according to the invention that in all operating areas it is possible to drive with slipping friction clutch and the friction clutch is controlled not dependent on slip but dependent on torque. The slip is then adjusted by itself and a slow slip regulation or adaption underlies the correction of the transfer torque. With switching processes the friction clutch bridging the converter is not opened but controlled in dependent on torque. For torque control a rising friction characteristic line is helpful wherein the friction value should preferably increase as the slip rises and the adhesive friction value should be less than the sliding

friction value.

Preferably within the scope of a further development the torque to be transferred by the friction clutch is detected in dependence on the torque of the drive assembly according to the torque equation

$$M_{Kupplung} = k_{me} \times k_{korr} \times (M_{Antriebsaggregat} + M_{korr\_MOT}) + M_{korr\_WÜ}$$

with

$M_{Kupplung}$	-	Torque on the friction clutch
k_{me}	-	Torque division factor ($0 \leq k_{me} \leq 2$)
k_{korr}	-	Correction factor for compensating multiplicative faults
M_{korr_MOT}	-	Correction torque for compensating faults adding to the engine torque
M_{korr}-WÜ	-	Correction torque for compensating faults adding to clutch torque

wherein a minimum slip is independently set between the drive and output of the torque transfer system in dependence on the size of the torque division factor k_{me} which is constant over the entire operating range of the drive train and any deviations from the ideal state are compensated long-term by the correction factor k_{korr} and the correction torques M_{korr_MOT} and $M_{korr\_WÜ}$.

With this development of the control method according to the invention a slip value of the friction clutch bridging the converter is produced which can be kept low by presetting the factors k_{me} and K_{korr}. In certain areas, such as at low speed and high load (where many internal combustion engines show a torque weakness) the factors are to be selected so that the torque to be transferred by the friction clutch is as small as possible so that a higher differential speed is set. Particularly in conjunction with a soft converter and a large conversion an increase in the output torque is then

reached in the particularly important operating areas which simulates a higher torque of the drive assembly.

The control method according to the invention is thus characterised by good vibration isolation with small slip, better reactions in the drive train during gear shift processes and load change processes as well as larger acceleration reserves, but also allows smaller and/or flatter torque converters which is of importance in the case of vehicles with front wheel drive and transversely mounted internal combustion engines. Finally there is an advantage in fuel consumption which should not be underestimated since with the method according to the invention the converter is bridged by the friction clutch in all gears. The torque distribution factor k_{me} of the torque equation given in patent claim 2 can be a value dependent on the output speed, on the speed of the drive assembly alone, both on the speed and on the torque of the drive assembly or even a value dependent both on the output speed and on the torque of the drive assembly. Also the speed of the drive machine is an important indicator for the factor k_{me}, namely either on its own or in conjunction with the torque released by the drive assembly.

For the construction and function of the torque transfer system and for the implementation of the method it is expedient if the friction clutch can be operated by pressurised flow medium and is designed so that two separate pressure chambers are formed between the friction clutch and converter cover and between the friction clutch and the rest of the converter housing, and a differential pressure existing between these pressure chambers determines the torque transferred by the friction clutch.

According to a further expedient development of the

invention in the case of a transfer system with an internal combustion engine as drive assembly its operating state can be determined in dependence on the engine speed and throttle valve angle, in dependence on the engine speed and on the inlet manifold underpressure or in dependence on the engine speed and injection time. With the alternatives given above the engine speed in conjunction with a further value, such as the throttle valve angle, inlet manifold underpressure or injection time, always serves as the indicator for the operating state.

As a result of the dynamic behaviour of the hydraulic and mechanical systems a too rapid increase in the amount of a parameter controlling the division of the torque between the converter and friction clutch to be transferred by the torque transfer system can lead to the generation of vibrations of a different frequency through too much jerk or sticking of the friction clutch.

In order to avoid such vibrations a suitable development of the invention proposes that the setting of an amount newly calculated and differing from the previous amount of a parameter affecting the division of the torque to be transferred between converter and friction clutch, preferably the differential pressure, is carried out delayed after a function in dependence on time.

The setting of an amount newly calculated and differing from the previous amount of a parameter affecting the division of the torque to be transferred between converter and friction clutch can however also be carried out delayed after a function in dependence on the differential speed between the output and drive of the torque transfer system.

Similarly the setting of an amount newly calculated and

differing from the previous amount of a parameter affecting the division of the torque to be transferred between converter and friction clutch is possible delayed after a function in dependence on the gradient of the engine speed.

When using a friction clutch which can be operated by pressurised flow medium it is possible according to a further development of the invention to regulate the differential pressure desired at the friction clutch by means of a PI or PID regulator wherein the control path from the differential pressure at the friction clutch, necessary to achieve a certain torque to be transferred by the friction clutch, to the adjusting differential pressure cannot be clearly analytically described.
It is however also possible to set the desired differential pressure at the friction clutch in that a pressure-proportional signal such as a valve flow is derived from a characteristic line and set whereby the compensation of any deviations occurring between the ideal and actual pressure is carried out by an I-return. As an alternative it is however also possible to set the desired differential pressure at the friction clutch in the way where a signal proportional to the desired differential pressure, such as a flow or scanning ratio, is calculated and regulated by means of a PI-I or PID regulator.

A further important variation of the method proposes that deviations of the torque actually transferred by the friction clutch from desired torque are determined by measuring the setting slip between the drive and output of the torque transfer system and comparing it with ideal values. Such deviations can however also be detected according to a further development where the torque transferred by the torque converter is calculated from its characteristic and thus the actual torque division between

the converter and friction clutch is checked. Finally differences occurring between the torque actually transferred by the friction clutch and the desired torque can also be attributed to multiplicative faults, to faults adding to the engine torque, to faults adding to the clutch torque, to faults multiplying and adding to the engine torque, to faults multiplying and adding to the clutch torque or to faults multiplying and adding to both the engine torque and clutch torque, and such faults are compensated with a time constant of several seconds in order to achieve only an adaptive character of the control.

A renewed method variation is characterised in that when a desire to accelerate is signalled on the part of the driver which is preferably notified by a change in speed of the throttle valve angle, the slip in the torque transfer system is increased by reducing the k_{me}-factor and thus the increased torque offered by the converter can be used as additional torque reserves.

Finally with a further method variation the slip in the torque transfer system is determined in all gears by the friction clutch whereby the degree of efficiency of the output transfer through the converter recedes and allows a converter interpretation with regard to a high stall speed and wide converter range. Thus the torque reserve which is available can be substantially increased with a deliberate increase in the torque transfer system.

A further method variation is characterised in that the slip in the torque transfer system is determined with all translations by the friction clutch whereby the degree of efficiency of the output transfer through the converter recedes and a converter interpretation is allowed with regard to a high stall speed and wide conversion range.

The problem of the invention regarding providing an improved lock-up clutch is solved by a clutch which can be operated by pressurised flow medium and which has a pump wheel, turbine wheel, guide wheel and a converter cover which is central with the axis of rotation, is connected rotationally secured with the pump wheel and encloses the turbine wheel, wherein a central ring piston mounted between the converter cover and turbine wheel is formed radially outwards as a conical clutch friction disc as described in further detail below. The ring piston can thereby have radially inside a sealing hub mounted on a counter sealing hub which is connected rotationally secured with the turbine wheel.

A further basic idea of the invention relates to a method for controlling a torque transfer system which is in active connection with the output of a drive assembly such as an internal combustion engine and is in driving connection through an output shaft with an automatic gearbox wherein the torque transfer system has a torque converter and a friction clutch which is mounted parallel thereto and can be operated by pressurised flow medium and has two pressure chambers mounted between a turbine wheel of the converter and a converter cover and designed so that a differential pressure existing between these pressure chambers determines the torque transferable by the friction clutch, and which is furthermore equipped with a measured value detecting system, a central computer unit and a hydraulic system which in conjunction with the computer unit produces a deliberate change in the differential pressure between the two pressure chambers and thus in the torque transferred by the friction clutch.

Processes for controlling torque transfer systems wherein the torque to be transferred by the friction clutch is set

by deliberately setting the differential pressure between the pressure chambers of a friction clutch mounted parallel with a converter and bridging same are known.

Thus DE-OS 3130 871 describes in connection with a torque transfer system of the above kind a regulating method wherein the ideal slip values occurring between the drive and output are measured, compared with predetermined ideal slip values and any detected differences are counter-adjusted. The latter is brought about by changing the difference between the pressurised flow medium biasing of the two pressure chambers of a friction clutch. This is thus a regulating method based on the standard slip regulation.

US-PS 5 029 087 also discloses a regulating method for converters having parallel mounted friction clutch wherein the slip at the clutch is measured, compared with predetermined ideal slip values and changed in dependence on the deviations detected in the differential pressure between the two pressure chambers of the friction clutch. Similarly here this is a typical slip regulation wherein measured deviations from the predetermined slip values are counteracted.

Finally US PS 4 577 737 also discloses a method for influencing a torque transfer system of the kind mentioned above wherein the torque transfer through a hydrodynamic converter is measured directly by a torque sensor and the torque transfer is fixed in dependence on the operating state of the drive machine. The end of the friction clutch bridging the converter is thereby adjusted so that the required torque transfer is to be guaranteed.

With this control method the torque transferred by the converter, like the adjusting slip, can obviously only then

be measured and influenced after it has been set. This is therefore also a regulation concept related to slip regulation although here working with the torque to be transferred by the converter.

Slip regulations of this kind where the difference between the output speed of a drive machine and the input speed of a gearbox mounted at the output side of the torque transfer system or a value corresponding to this speed difference is measured, compared with ideal values and any possible deviation between the actual and ideal values is counteracted have not proved completely satisfactory.

Thus when changing gear the speed difference changes as a result of changes in the torque. The speed regulation thereby takes place so late that it can lead on the output side or in the gearbox torque to undesired over vibration. Furthermore when switching at the end of a switching process it can result in sticking of the friction clutch bridging the converter. Consequently the friction clutch must be opened with all switching processes. The slip regulation tries during switching processes to keep the speed difference between the output speed of the drive machine and the input speed of the gearbox to the ideal level, thus works against the gearbox mounted at the output side of the torque transfer system.

Consequently the object of the invention is to provide an improved method for controlling a torque transfer system which has a converter, a friction clutch bridging same and an automatic gearbox at the output side, and to provide at least in connection with other inventive ideas of the present application improved mechanical components which can be used in a particularly advantageous manner, such as an improved converter and an improved friction clutch.

This problem underlying the invention regarding the control method is solved in that with the control method according to the preamble of patent claim 1 the torque to be transferred by the friction clutch is detected in dependence on the operating state of the drive assembly according to the torque equation

$$M_{Kupplung} = k_e \times k_{korr} \times M_{Antriebsaggregat}$$

with $k_e = k_{me}$ as the torque division factor and k_{korr} as the correction factor

and the force biasing of the friction clutch which is required to transfer the predetermined clutch torque is calculated and adjusted wherein the slip is independently set between the drive and output of the torque transfer system in dependence on the size of the torque division factor k_e and the correction factor k_{korr} compensates any deviations of each special drive train from the ideal state.

A further inventive basic idea relates to a method for controlling a torque transfer system which is in active connection with the output of a drive assembly, possibly an internal combustion engine, and is in driving connection through an output shaft with an automatic gearbox wherein the torque transfer system has a torque converter and a friction clutch mounted parallel thereto, a measured value detection system and a central computer unit wherein the force biasing of the friction clutch and thus the torque transferred by same can be consciously changed in conjunction with a central computer unit, wherein the torque to be transferred by the friction clutch is detected in dependence on the operating state of the drive assembly according to the torque equation

$$M_{Kupplung} = k_e \times k_{korr} \times M_{Antriebsaggregat}$$

with $k_e = k_{me}$ as the torque division factor and k_{korr} as the correction factor

and the force biasing of the friction clutch which is required to transfer the predetermined clutch torque is calculated and adjusted wherein the slip is independently set between the drive and output of torque transfer system in dependence on the size of the torque division factor k_e which is constant over the entire operating range of the drive train and the correction factor k_{korr} compensates any deviations of each special drive train from the ideal state. The invention likewise relates to a method for controlling a torque transfer system which is in active connection with the output of a drive assembly, possibly an internal combustion engine, and is in driving connection through an output shaft with an automatic gearbox wherein the torque transfer system has a torque converter and a friction clutch mounted parallel thereto, a measured value detection system and a central computer unit wherein the force biasing of the friction clutch and thus the torque transferred by same can be consciously changed in conjunction with a central computer unit, wherein the torque to be transferred by the friction clutch is detected in dependence on the operating state of the drive assembly according to the torque equation

$$M_{Kupplung} = k_e \times k_{korr} \times M_{Antriebsaggregat}$$

with $k_e = k_{me}$ as the torque division factor and k_{korr} as the correction factor

and the force biasing of the friction clutch which is required to transfer the predetermined clutch torque is

calculated and adjusted wherein the slip is independently set between the drive and output of the torque transfer system in dependence on the size of the torque division factor k_e which is independent of the engine characteristic field and the correction factor k_{korr} compensates any deviations of each special drive train from the ideal state.

The object underlying the invention can also be solved by a method for controlling a torque transfer system which is in active connection with the output of a drive assembly, possibly an internal combustion engine, and is in driving connection through an output shaft with an automatic gearbox wherein the torque transfer system has a torque converter and a friction clutch mounted parallel thereto, a measured value detection system and a central computer unit wherein the force biasing of the friction clutch and thus the torque transferred by same can be consciously changed in conjunction with a central computer unit, wherein the torque to be transferred by the friction clutch is detected in dependence on the operating state of the drive assembly according to the torque equation

$$M_{Kupplung} = k_e \times k_{korr} \times M_{Antriebsaggregat}$$

with $k_e = k_{me}$ as the torque division factor and
k_{korr} as the correction factor

and the force biasing of the friction clutch which is required to transfer the predetermined clutch torque is calculated and adjusted wherein the slip is independently set between the drive and output of the torque transfer system in dependence on the size of the torque division factor k_e which is dependent on the speed of the drive assembly alone and the correction factor k_{korr} compensates any deviations of each special drive train from the ideal state.

A further solution to the problem exists in a method for controlling a torque transfer system which is in active connection with the output of a drive assembly, possibly an internal combustion engine, and is in driving connection through an output shaft with an automatic gearbox wherein the torque transfer system has a torque converter and a friction clutch mounted parallel thereto, a measured value detection system and a central computer unit wherein the force biasing of the friction clutch and thus the torque transferred by same can be consciously changed in conjunction with a central computer unit, wherein the torque to be transferred by the friction clutch is detected in dependence on the operating state of the drive assembly according to the torque equation

$$M_{Kupplung} = k_e \times k_{korr} \times M_{Antriebsaggregat}$$

with $k_e = k_{me}$ as the torque division factor and
k_{korr} as the correction factor

and the force biasing of the friction clutch which is required to transfer the predetermined clutch torque is calculated and adjusted wherein the slip is independently set between the drive and output of the torque transfer system in dependence on the size of the torque division factor k_e which is dependent on both the speed and the torque of the drive assembly and the correction factor k_{korr} compensates any deviations of each special drive train from the ideal state.

With the invention it is therefore a question of dividing the input torque into a hydraulic part which is to be transferred by the converter and a mechanical part which is to be transferred by the friction clutch. For the infinite

control the lock-up clutch is biased with variable force which is controlled by an intelligent control system so that an optimum division of converter torque and lock-up torque is produced for each relevant driving situation.

For the construction and functioning of the torque transfer system or for implementing the method it can be expedient if the friction clutch can be operated by pressurised flow medium and is designed so that two separate pressure chambers are formed between the friction clutch and converter cover and between the friction clutch and the remaining converter housing and that a differential pressure existing between these pressure chambers determines the torque which can be transferred by the friction clutch.

A characteristic of the control method according to the invention is thus that in all operating areas it is possible to drive with slipping friction clutch and the friction clutch is controlled not dependent on slip but in dependence on moment. The slip is then set by itself and a slow slip regulation is provided to correct the transfer torque. During gear change processes the friction clutch bridging the converter is not opened but responds dependent on torque. A rising friction characteristic line is helpful for the torque control wherein the friction value should preferably increase as the slip rises and the sticking friction value should be less than the slide friction value.

With the control method according to the invention a slip value of the friction clutch bridging the converter is produced which can be kept small by predetermining the factors k_e and k_{korr}. In certain areas, possibly at low speed and high load (where many internal combustion engines show a torque weakness) the factor is selected so that the torque to be transferred by the friction clutch is so low that a

higher differential speed is set. Particularly in conjunction with a soft converter and a large conversion an increase in the output torque is then achieved in the particularly important operating areas which simulates a higher torque of the drive assembly.

The control method according to the invention is thus characterised by good vibration isolation with little slip, better reactions in the drive train during gear change processes and load change processes as well as greater acceleration reserves but also allows smaller and/or flatter torque converters which is of significance in the case of vehicles with front wheel drive and transverse mounted engines. Finally there is the advantage of consumption which should not be underestimated since with the method according to the invention the converter is bridged by the friction clutch in all gears.

According to an expedient development of the invention with a transfer system with an internal combustion engine as the drive assembly its operating state can be determined in dependence on the engine speed and on the throttle valve angle, in dependence on the engine speed and inlet manifold underpressure or in dependence on the engine speed and injection time. With the aforesaid alternatives the engine speed in connection with a further value such as the throttle valve, inlet manifold underpressure or injection time always serves as the indicator for the operating state.

Another expedient development of the invention proposes that the factor k_e of the torque equation given in patent claim 1 is a value which is constant over the entire operating area of the drive train and is dependent on the speed of the drive assembly alone, or both on the speed and on the torque of the drive assembly. Also the speed of the drive machine

is an important indicator for the factor k_e, namely either by itself or in connection with the torque released by the drive assembly.

According to another embodiment the control method according to the invention can also be characterised in that a torque to be transferred by the friction clutch and detected in the central computer unit in dependence on a torque change in the drive train and which (torque) differs from the momentary torque is set by predetermining the value of any parameter X determining the torque to be transferred by the friction clutch and which (value) is desired after a scanning interval at time point tn+1, after a function which excludes undesired incidents, such as eg sticking of the friction clutch, by calculating the gradient ΔX which is required to obtain the desired value of the parameter X after a time interval Δt, by setting the calculated gradient ΔX by means of the hydraulic system and by repeating the above sequence until an ideal value X_{Soll} is reached.

In particular the method according to this embodiment can be characterised by a proportionality regulation wherein as parameter the differential pressure ΔP between the pressure chambers of the clutch is predetermined according to the equation

$$\Delta P_{n+1} = (1 - ß) \times \Delta P_{Soll} + ß \times \Delta P_n$$

with

$$ß = f(T_v, t).$$

As an alternative a new value of the torque transferred by the friction clutch and detected in the central computer unit in dependence on a torque change in the drive train can

be set by calculating the gradient ΔX of any parameter X determining the torque to be transferred by the friction clutch after a function which excludes undesired incidents, such as eg temporary sticking of the friction clutch, by setting the desired gradient ΔX by means of the hydraulic system and by repeating the sequence until the required ideal value X_{Soll} is reached. With this alternative the gradient of the pressure difference P between the pressure chambers of the clutch can be calculated as the parameter according to the equation

$$\Delta\Delta P = C_1 \times (\Delta P_{Soll} - \Delta P_n).$$

This means:

$$\Delta\Delta P = C_1 \times (\Delta P_{Soll} - \Delta P_{Ist})$$

(see also Figure 10).

ΔP.....	Change in the differential pressure ΔP in the next time interval
ΔP_{Soll}..	Ideal pressure difference
ΔP_n.....	Actual pressure difference at time t_n
C_1......	Proportionality or amplification factor with $0 \leq C_1 \leq 1$

The amplification factor C_1 determines how quickly a deviation between ΔP_{Soll} and ΔP_n is compensated.

Boundary values: $C_1 = 0$, $C_1 = 1$.

With $C_1 = 0$ no compensation takes place since the pressure increase $\Delta\Delta P$ in the next computer interval would be equal to nought.

$C_1 = 1$ is equal to an ideal value jump since the entire deviation between ideal and start value (ΔP_{Soll}, ΔP_{Start} in

Figure 10) would have to be completed within a time interval. These two boundary values thus only have a theoretical value. The important area is $0 < C_1 < 1$. This affects how quickly a deviation takes place between ideal and actual value. The smaller C_1, the longer it takes for compensation.

The advantage of this type of compensation of a deviation between ideal and actual value lies in the fact that with a large deviation between ideal and actual value a large setting value, that is a large value for $\Delta\Delta P$ is calculated. If the ideal value and actual value come close then the value for $\Delta\Delta P$ becomes smaller and smaller and a "soft" merge between the ideal and actual values occurs. This counteracts any generation of vibrations.

An advantageous processing variation proposes that in operating conditions where a reduction in the input torque at the torque transfer system is to be expected, such as when changing translation or when switching in additional assemblies, a possible temporary sticking of the friction clutch through the reduction in the torque transferred by the friction clutch is counteracted in that either the torque division factor k_e or the correction factor k_{korr} is reduced by a predetermined amount and is raised again after a function in dependent on time to a value which is optimum for the vibration insulation and fuel economy.

Likewise within the scope of the control method according to the invention in operating conditions where a reduction in the input torque at the torque transfer system is to be expected, such as when changing back down gear or when switching in additional assemblies a possible temporary sticking of the friction clutch through a reduction in the torque transferred by the friction clutch can be

counteracted in that either the torque division factor k_e or the correction factor k_{korr} is reduced by a predetermined amount and is raised again after a function in dependence on time to a value which is optimum for the vibration insulation and fuel economy.

Another method variation provides that the correction factor k_{korr} compensates deviations of each special drive train from the ideal state in that preferably in a fixed quasi stationary area the setting slip is measured and compared with ideal slip values which guarantee an optimum vibration isolation with the highest possible fuel economy, and the factor k_{korr} is compensated in the event of a deviation between ideal and actual slip values.

Another method variation is characterised in that when a desire for acceleration is signalled on the part of the driver, which is preferably notified by a change in the speed of the throttle valve angle, by reducing one of the factors k_e or k_{korr} the slip in the torque transfer system is increased and the torque increase offered by the converter can thereby be used as additional torque reserves.

Finally with a further method variation the slip in the torque transfer system is preferably determined in all gears by the friction clutch whereby the degree of efficiency of the output transmission through the converter recedes and allows a converter interpretation regarding the widest possible conversion area with which the torque reserve available can be significantly increased with a deliberate increase in the slip in the torque transfer system.

It can likewise be advantageous if the slip in the torque transfer system is determined with all translations by the friction clutch whereby the degree of efficiency of the

output transmission through the converter recedes and a converter interpretation is possible regarding the widest possible conversion area.

A further basic idea of the invention relates to a torque transfer system for the drive train of a vehicle equipped with gear change gearbox, more particularly a motor vehicle with internal combustion engine drive, with a torque converter, which is in driving connection with a drive assembly of the vehicle and is in active connection through an output shaft with an automatic gearbox connected at the output side, with a friction clutch mounted parallel with the hydrodynamic converter which (clutch) can be operated by pressurised flow medium and has a pressure chamber mounted between a turbine wheel of the converter and a ring piston actively connected to a friction disc on one side, and between the ring piston and a converter cover on the other side wherein the pressure chambers are designed so that a differential pressure existing between these pressure chambers determines the torque transferable by the friction clutch, with a measured value detection system, a central computer unit and with a hydraulic system which provides in conjunction with the computer unit a deliberate change in the differential pressure between the two pressure chambers and thus the torque transferable by the friction clutch.

Torque transfer systems where the torque to be transferred by the friction clutch is set by deliberately setting the differential pressure between the pressure chambers of the friction clutch which is mounted parallel to a converter and bridges same are known.

Thus in the DE-OS 31 30 871 already mentioned a torque transfer system of the above kind is described wherein the slip values arising between the drive and output are

measured, compared with predetermined ideal slip values and any differences detected are counteracted. This takes place by changing the difference between the pressurised flow medium biasings of the two pressure chambers of a friction clutch mounted parallel to a hydrodynamic converter.

From the US-PS 5 029 087 similarly already mentioned a torque transfer system with converter and parallel mounted friction clutch is also already known wherein the slip is measured at the clutch, compared with predetermined ideal slip values and the difference between the two pressure chambers of the friction clutch is changed in dependence on the deviations detected.

Finally from US PS 4 577 737 a torque transfer system of the above mentioned kind is also known wherein the torque transfer through the hydrodynamic converter is measured directly by means of a torque sensor and the torque transfer is fixed in dependence on the operating state of the drive machine. The end of the friction clutch bridging the converter is thereby regulated so that the required torque transfer is guaranteed.

Characteristic of the torque transfer systems according to the prior art is that the friction clutch which is mounted parallel to the torque converter and which is completely open in the lower gears becomes switched on in the upper gears. In order to achieve a good overall degree of efficiency and to restrict the build up of heat the converters are designed "hard". In view of this "hard" converter design the torque increase drops sharply as the speed increases with the result that in the middle speed area only a very restricted torque increase takes place and in the upper speed area none at all takes place.

The object of the present invention is therefore to provide a significantly improved torque transfer system where in the interest of acceleration reserves in the middle and higher speed areas, an effective torque increase is still reached and the fuel consumption can be reduced.

This is achieved according to the invention in that with the torque transfer system according to the preamble of claim 43 the friction clutch is controlled in all driving gears and that the torque converter has compared to conventional converters a higher conversion which is preferably greater than 2.5. It can be particularly expedient if the torque conversion between the turbine wheel and pump wheel is in the order of 2.5 to 3.5.

This object can also be achieved if the friction clutch is controlled with all translations and that the torque converter has a torque conversion of preferably more than 2.5. It can be particularly advantageous if the torque conversion is substantially in the area of 2.5 to 3.5.

With a predetermined internal combustion engine it is particularly expedient if the torque converter which is used in connection with the invention has a smaller capacity factor than the torque converter used for this engine. This means that with a torque path predetermined by the internal combustion engine the fixed braking speed of the torque converter according to the invention is higher than with a conventional converter. By fixed braking speed is meant the speed at which the path of the torque taken up by the turbine wheel intersects with the torque characteristic line of the internal combustion engine. In order to detect this speed the turbine wheel is blocked and the pump wheel is driven by the internal combustion engine. With the designs of the torque converters up until now the fixed braking

speed is in the order of 1800 to 3000 revolutions per minute. Through the design according to the invention this fixed braking speed can even be moved into the area above 300 rpm. The smaller the capacity factor so the softer the converter. This also means that the path of the turbine or pump torque over the turbine and pump speed is flatter compared to the converters used hitherto.

With the invention the converter is thus designed "soft" and can also have a substantially wider secondary characteristic field.

Greater acceleration reserves are thereby available which can be used in particular in overtaking or accelerating phases and often make changing down to a lower gear unnecessary.

The additionally usable area of the secondary characteristic field of the converter designed according to the invention is mainly used only in non-stationary conditions. The amount of heat building up at this time is no higher than with conventional systems and therefore non-critical. It has proved equally suitable according to a further embodiment of the invention if with the torque transfer system the heat building up during travel is calculated up by the computer unit and the actual heat balance thus produced is compared with the amount of heat which the design structure allows. The oil temperature is moreover measured so that calculations can start from the actual temperature level.

Through this measure an unusually high heat build-up is recognised in good time and thus the prerequisite for a reduction in the amount of heat is provided. If the heat strain on the overall system is too great then the slip is

reduced. If the strain on the friction face is too great then the slip is changed in dependence on the driver's wishes: If the driver wants to accelerate and conversion can still be offered then the lock-up torque is reduced and thus the slip increased. Otherwise the lock-up torque is increased and thus the slip is reduced.

Another important development of the invention proposes that a damper unit acting between the turbine of the converter and the friction disc of the lock-up clutch is preferably designed for the partial load area in which a complete converter lock-up comes into consideration. This allows a substantially better damping of rotary vibrations than in the case of conventional dampers which are designed for full load. In the remaining area the isolation of high-frequency vibrations through the slip is guaranteed.

The torsion damper dampens or filters the vibrations and torque irregularities transferred by the lock-up clutch at least to an acceptable extent wherein the stop torque of the torsion damper, thus the maximum torque which can be transferred by the damping means, such as springs, is smaller than the nominal torque, thus the maximum torque of the internal combustion engine. This means that according to the invention the torsion damper is not as with the previously known prior art designed for the full load of the drive assembly or internal combustion engine. As soon as the stop torque is reached the lock-up clutch or the torsion damper thereof behaves in practice as a rigid drive member in the rotary drive direction. Since the torsion damper according to the invention for a lock-up clutch of a hydrodynamic torque converter is only designed for a partial load area this torsion damper can be constructed particularly simply whereby a cost-effective production is also ensured. Furthermore the energy accumulators of the

torsion damper, such as in particular coil springs, can be made weaker so that these also require less structural space whereby the structural space required for the lock-up clutch or torsion damper can likewise be reduced. Furthermore there is a saving in weight. In order to protect the energy accumulators of the torsion damper against overloading it is preferable if special stops are provided between the input part and output part of the torsion vibration damper of the lock-up clutch.

This can be achieved if the friction clutch or torque transfer system is controlled so that in all forward gears or with all forward translations a partial closing of the lock-up clutch takes place at least at times.

For most cases it has proved expedient if the stop torque or lock-up torque of the torsion damper is in the order of between 10 and 60% of the maximum, thus nominal torque of the combustion engine, preferably in the order of 25 to 50%. In many cases however the lock-up torque or stop torque of the torsion damper can also have larger or somewhat smaller values. According to a further development of the invention a torsion damper designed in this way for a lock-up clutch has no special friction device. This means that between the input part and output part of the torsion damper there are only energy accumulators which oppose a relative rotation between these parts.

Through the design according to the invention of the torque transfer capacity of the torsion damper the vibrations occurring in the partial load area, thus in the area with drive torques in the order of between 10 and 60 %, or between 25 and 50% of the nominal torque, can be damped with very good effect.

It can be particularly expedient if the damper allows an angle of travel which is relatively small compared to the angles of travel known hitherto of the dampers for converter lock-up clutches. This angle of travel can be in the order of ± 2 to ± 8°, preferably in the order of ±3 to ±6°. The overall angle of travel of the damper, thus the overall angle of travel for both turning directions thus amounts to 4 to 16°, preferably 6 to 12°. As a result of the comparatively small angle of travel of a torsion damper according to the invention for lock-up clutches it can be guaranteed that with a change of load, thus when changing from push-type operation to pull-type operation and vice versa the deflections in the damper are kept small whereby a rocking of the drive train can be restricted or avoided. Advantageously the torque shocks or torque proportions of these shocks which lie above the stop torque of the torsion damper can be damped or filtered through slip or slipping of the lock-up clutch so that they are kept substantially away from the output train or gearbox.

For most cases of use it can be expedient if the damper has a rotational stiffness which is in the order of between 7 and 30 Nm/°, preferably between 8 and 15 Nm/°. In many cases however this rotational stiffness can also be selected smaller or larger. In most cases the lock-up clutch or torsion damper can be designed so that this has a stop torque in the order of between 30 and 90 Nm, preferably in the order of between 40 and 70 Nm. For low powered vehicles the stop torque can however also be made smaller. Similarly it may be necessary in the case of high powered vehicles with comparatively high weight to make the stop torque greater.

The drive system according to the invention can

advantageously be used in conjunction with a method for controlling a lock-up clutch which is slip-controlled in dependence on the ensuing torque and which ensures a control designed from the energy and performance-related points of view at least in all forward gear phase of a gearbox. The drive system according to the invention can however also be used in connection with gearbox controls or regulations which leave the lock-up clutch completely open in the first and/or second forward gear stage or stages.

According to a further possibility according to the invention of a design of a drive system or torque transfer system of the kind already mentioned the torque control or torque regulation of the lock-up clutch of a hydrodynamic torque converter can be carried out so that the lock-up clutch has at least two operating ranges in which the adjustment of the size of the torque transferable by the lock-up clutch in relation to the ensuing torque of the drive machine takes place according to other points of view or another mode. Thus at least one of the correction factors k_{me} (torque division factor), K_{korr} (correction factor for compensating multiplicative faults), MkorrMot (correction torque for compensating faults adding to the engine torque) and MkorrWÜ (correction torque for compensating faults adding to the clutch torque) is evaluated differently in the two operating areas. This means that the size of at least one of these factors, preferably the k_{me} -factor and thus also the effect of this size on the torque transferable by the lock-up clutch is defined differently in the two areas. It can be particularly advantageous if in a first area the torque transferable by the lock-up clutch is in the order of between 10 and 60%, preferably between 15 and 50% of the maximum torque of the drive machine, such as in particular internal combustion engine and in the adjoining second area

the torque transferable by the lock-up clutch lies above the upper torque boundary value of the first area, ie is thus greater than 50 to 60% of the maximum torque of the combustion engine. It can be particularly expedient if the maximum torque transferable in the first operating area through the lock-up clutch agrees at least substantially with the stop torque of the torsion damper of the lock-up clutch. Through such a design it is ensured that torque vibrations with smaller amplitudes are absorbed or filtered by the torsion damper whilst vibrations with torque peaks lying above the stop torque of the torsion damper can be at least substantially damped through slipping of the lock-up clutch.

The torque regulation or torque control of the lock-up clutch in the first area can advantageously be carried out so that the torque transferable by the lock-up clutch at least substantially over the entire first area is greater than the relevant ensuing torque of the internal combustion engine which is produced by this as a result of the amount of fuel supplied to same. The torque transferable by the lock-up clutch can thereby be set over the first speed range so that this torque is changed at least over a significant area of the first speed area approximately in synchronization with the torque change in the internal combustion engine in the first area. This means that with a reduction in the torque released by the combustion engine the torque transferable by the lock-up clutch also decreases whereby this remains greater however than the torque of the combustion engine. With an increase in the torque released by the internal combustion engine the torque transferable by the lock-up clutch becomes correspondingly greater. It can thereby be expedient if the torque transferable by the lock-up clutch amounts to 1 to at least 1.2 times the relevant ensuing engine torque of the combustion engine.

According to a further embodiment of the invention the torque transferable in the first area by the lock-up clutch can be set at least approximately to a constant value wherein this value can lie in the order of between 25 and 60% of the maximum torque of the internal combustion engine, preferably in the order of 30 to 50 % of the maximum torque. Advantageously this value can correspond at least approximately to the stop torque or lock-up torque of the torsion damper of the lock-up clutch, preferably however can be larger, eg 1.05 to 1.2 times this lock-up torque.

According to another suitable embodiment the setting of the torque transferable by the lock-up clutch in the first speed range can also be carried out so that in a lower partial area of this first area which advantageously adjoins the idling speed of the internal combustion engine, the torque transferable by the lock-up clutch is kept substantially at a constant value and in the adjoining second partial area of this first area the torque transferable by the lock-up clutch follows the torque development of the internal combustion engine. The latter means that if the torque of the internal combustion engine becomes greater in the second partial area the transferable torque of the lock-up clutch also becomes greater and vice versa. In this second partial area the torque transferable by the lock-up clutch is at least of the same size, preferably somewhat larger than the relevant ensuing torque of the combustion engine.

In order to ensure an accurate control or regulation of the torque transferable by the lock-up clutch it can be particularly advantageous if the torque transferable by the lock-up clutch in the first speed range does not drop below 1% of the nominal torque of the internal combustion engine and is preferably kept greater than 1% of this nominal

torque. A minimum pressure is thereby guaranteed for the lock-up clutch which can still be set satisfactorily with the known valves. As a result of the minimum pressure level this pressure can thus be kept within comparatively narrow limits.

For most cases it can be advisable if the first area extends from idling speed to a maximum of 3000 rpm, preferably up to a maximum value between 2000 and 2500 rpm. In many cases it can however also be expedient if the upper value lies above 3000 rpm or below 2000 rpm.

Within the scope of the invention the torque transferable by the lock-up clutch, seen over the entire operating range of the drive system, can be carried out so that in a first lower area of this overall operating range the vibration uncoupling is carried out at least substantially through the damper and in a second adjoining area the vibration uncoupling is ensured substantially by setting the slip in the lock-up clutch. In this second area the existing damper can additionally be brought into action at times, ie the energy accumulators of the damper can be relaxed and compressed again whereby however this damper has a secondary role in this second area regarding vibration uncoupling.

The torsion damper of the lock-up clutch designed substantially for the first speed range preferably has, as already mentioned a stop or lock-up torque which is in the order of between 10 and 60%, preferably between 15 and 50% of the maximum torque of the internal combustion engine. The torsion damper can however according to a further design possibility of the invention also be designed so that it still has adjoining the angle of travel which corresponds to the aforesaid torque value, a comparatively small angle of travel in which the spring rate amounts to a multiple or is

very steep so that the torsion damper has a designated stop suspension which prevents the component parts which provide the turning restriction in the torsion damper from striking each other too hard. The stop noises which may arise can thereby be substantially reduced. The ratio between the angle of travel allowed by the stop suspension and the remaining angle of travel connected in at the input can advantageously be in the order of 1 to 2 to 1 to 5, preferably in the order of 1 to 2.5. The rotational stiffness caused by the stop suspension is advantageously 4 to 10 times greater than the rotational stiffness of the torsion damper connected in at the input side of the stop suspension. Advantageously the end stop torque of the torsion damper caused by the stop suspension can amount to 2 to 5 times the aforesaid torque existing at the end of the first range. Advantageously however the maximum torque which can be transferred by the stop suspension is less than the maximum engine torque. The angle of travel covered by the stop suspension between the input part and output part of the torsion damper can be in the order 0.5 to 3°, whereby it can be advantageous if this angle lies in the order of 1 to 2°. The stop suspension can also be designed so that this is only active in the pulling direction.

By designing the torsion vibration damper for a lock-up clutch according to the invention it is possible to remove the aforesaid problems of humming which occur with a comparatively low torque. This is probably due to the fact that the said sticking phases of the clutch are bridged by the rotationally elastic torsion damper.

According to another suitable development of the invention the torque transferable by the lock-up clutch at least in the first area can be reduced when conditions occur with high vibration amplitude in the drive train, that is for

example in the case of resonance, load change impact or the like, whereby the slip in the lock-up clutch is increased. In the event of load change impact the torque transferable by the lock-up clutch can, if necessary, be broken down practically completely in the pushing phase. A reduction in the torque transfer capacity of the lock-up clutch in the case of the aforesaid operating conditions can also advantageously take place in the second speed range.

According to a preferred embodiment the drive system or the transmission system can be designed so that at least the major part of the characteristic field of the internal combustion engine used in the main driving area falls within the first area. This main driving area can advantageously include at least the areas of the engine characteristic field which are relevant for the FTP75-cycle and/or for the ECE cycle town, open road and motorway traffic (town 90 km/h, 120 km/h). Through such a design it is ensured that in the main driving area the vibration isolation takes place in practice mainly over the damper and thus the converter is practically always bridged whereby an energy-saving and fuel-saving operating procedure is guaranteed. This is not the case with the previously known drive systems with slipping lock-up clutch since with these a slip is just adjusted in the first speed area, as is apparent from the prior art already mentioned. Since according to the invention the torsion damper is preferably designed for the main driving area a substantially better damping of the rotary vibrations occurring there can be achieved than would be possible with a damper design intended for a larger driving area. Furthermore a particularly compact converter construction is produced.

According to a design possibility of the invention in the second speed area the torque transferable by the lock-up

clutch can amount to 0.6 to about 1 times the relevant ensuing torque of the internal combustion engine, preferably 0.8 to 0.9 times. It is expedient if the torque transferable by the lock-up clutch in the second speed range always remains below the ensuing engine torque. Through such a design it can be guaranteed that in the second operating area there is always a slight slip in the lock-up clutch which serves to dampen the torque irregularities which occur there and which cause torsion vibrations.

With non-critical vehicles, ie with vehicles which in the second speed range or operating range have no greater irregularities in the torque release the lock-up clutch can also be practically closed which means that the torque transferable by the lock-up clutch corresponds at least to the torque released by the internal combustion engine at the corresponding time point, and preferably lies slightly above same. The ratio can thereby preferably lie in the order of between 1 and 1.2.

With the previous description there is always mention of two operating areas, whereby areas are meant which adjoin the idling speed. The invention is however not restricted to embodiments where the entire speed-relevant operating area of the drive system above the idling speed is only divided into two areas but is also directed to embodiments where the entire operating area is divided into more than two areas. Thus in many cases it can be expedient if the two areas already described are adjoined by a third area wherein in this third area a complete converter lock-up is always present. This third area thus comprises a speed span which lies above that of the second area, the lower value of this third speed area must thereby be fixed so that above this value no harmful excitations can be produced through the internal combustion engine so that no vibration damping is

required through slip.

According to a further development of the invention with a transmission system with an internal combustion as the drive assembly a device can be provided which at least with acceleration processes determines whether by opening the lock-up clutch and keeping the same gear or the same translation it is possible to achieve an increase in pulling power through torque conversion whereby in this case the lock-up clutch is opened and the engaged gear or the set translation is retained, otherwise the gearbox is shifted back at least one gear or the translation is changed whereby the clutch can then likewise be at least partially opened so that an increase in slip in the lock-up clutch takes place. The said device can be formed by an electronic computer unit or a processor which receives the required values or parameters through corresponding sensors. Many of these parameters can however also be stored in the electronic unit in the form of files or characteristic fields. Thus for example the characteristic field of the internal combustion engine and/or the characteristic field of the converter and/or the characteristic field of the converter lock-up clutch can be recorded in the electronic unit. The operating state of the internal combustion engine can furthermore be determined in dependence on the speed, throttle valve angle or on the amount of fuel supply, inlet manifold underpressure and if required on the injection time.

The invention generally relates to torque transfer systems and processes for same which are mounted in the drive train with an automatic gearbox. An automatic gearbox can thereby be a stepped gear with discrete translation or discrete translations or even an infinite gear with a controllable continuous translation, such as eg with an infinitely

adjustable cone pulley belt contact gearbox.

As already mentioned with the drive system according to the invention a complete lock-up of the converter can take place from a certain engine speed or a certain speed of the vehicle since above this engine speed a drive system which is practically rigid as a result of complete lock-up is substantially insensitive to the torsion vibrations which occur there. Thus above this specific engine speed the lock-up torque of the clutch can be set to a value which corresponds approximately to the engine torque or lies above same.

Through the design of the torsion damper according to the invention in conjunction with the control or regulating strategy for the torque transferable by the lock-up clutch the torque impulses which are produced in the partial load area of the internal combustion engine at the friction faces of the lock-up clutch and are due to the transitions between the sticking and sliding states and in turn can produce humming noises on the vehicle can be at least substantially reduced. Furthermore in this first area as a result of the low lock-up torque of the clutch set no jolting vibrations can build up. The softness of the torsion damper must be matched to each drive system or to each vehicle. If the torsion damper has a resonance area which must be passed through in the operation of the vehicle, then as soon as this area appears it is possible to allow a slip in the clutch. Humming or rattling can thereby be prevented.

Restricting the load change in the first area is not only produced by the small angle of travel of the torsion damper but also by the response of the lock-up clutch to a torque which lies at a comparatively low level in relation to the maximum torque of the internal combustion engine. As

already mentioned, the response can be produced in that at least in the first operating area the torque transfer capacity of the lock-up clutch lies only slightly above the ensuing engine torque. A vibration excitation of the drive train through load change vibrations can thus be substantially prevented by the design of the drive system according to the invention. In the second speed area which corresponds to a higher load on the combustion engine, bridging is with a smaller torque than the ensuing engine torque, and thereby results in slip. This slip likewise acts to absorb noise in a certain torque area, particularly in conjunction with the torsion damper since in this area it can still lead to sticking/sliding transitions between the friction faces of the lock-up clutch.

In the entire operating area or in the entire characteristic field area of the drive machine, such as in particular internal combustion engine, bridging is only effected if it appears expedient for energy reasons. There are namely areas where it is more advisable to drive non-bridged instead of partially or fully bridged. Also when the driver wishes to accelerate the lock-up clutch is opened in order to produce a torque conversion.

The drive system according to the invention and/or the method steps according to the invention for adjusting the torque transferable by the lock-up clutch can advantageously be used in conjunction with a soft torque converter. The use of such a soft converter allows a better acceleration behaviour in the case of motor vehicles since a converter of this kind has a greater torque conversion and thus a greater conversion area can be used. Furthermore the better degree of efficiency of the soft converter in the wide areas compared to conventional converters can be utilized whereby the output loss and thus consumption as well as the oil

temperature can be reduced. The area of the poorer performance of the soft torque converter is bridged or jumped, namely in that the converter lock-up clutch is closed in relation to the ensuing engine torque to a torque value which allows a certain slip. Through such a control or regulation of the converter or its lock-up clutch it can be ensured that in all driving conditions it is possible to drive with better performance and with lower lost power. Since through the design of the drive system according to the invention bridging is also possible in all driving stages and in all translations of the gearbox, the fuel consumption of the motor vehicle equipped with such a drive system can be substantially reduced to the level of a motor vehicle with converter-free or conventional gear shift.

This measure allows a particularly compact converter construction wherein the degree of efficiency from the point of view of the aforementioned lock-up control is of only secondary importance.

Further inventive measures are produced from the sub-claims, the description of the drawings and the drawings themselves.

A further basic idea of the invention relates as already mentioned to a lock-up clutch for a hydrodynamic torque converter with a pump wheel, turbine wheel, guide wheel and a converter cover which is central with the axis of rotation, is connected rotationally secured with the pump wheel and encloses the turbine wheel wherein according to a further independent inventive feature or one which can be used in combination with at least one further feature underlying the present invention, the central ring piston mounted between the converter cover and turbine wheel is provided radially outside with a conical clutch friction face. The ring piston can thereby have radially inwards a

sealing hub which is mounted on a counter sealing hub connected rotationally secured with the turbine wheel, and at least one ring-shaped damper element of a damper unit can be housed circumferentially between a damper output part connected rotationally secured with the ring piston and a damper output part connected rotationally secured with the turbine wheel.

The damper unit can thereby as already mentioned comprise a damper with rotationally resilient means which are designed ring-shaped and are mounted on the side of the ring piston pointing towards the converter cover between the hub part of the piston and the friction face interacting with a suitably conically designed counter friction face of the converter lid.

A lock-up clutch of the aforesaid kind with cones opening to the side pointing away from the turbine wheel has a particularly small axial structural length and also allows a spring damper to be mounted with large turning angles since the ring-shaped damper element can be mounted between the radially outer area of the turbine wheel and the clutch friction disc of the ring piston provided with a friction face. This leads to an increase in the gusset between the peripheral area of the turbine wheel and the clutch friction disc of the ring piston and thus to improved installation possibilities for the damper unit.

For many cases it can however also be advantageous if the interacting friction faces of the ring piston and the converter cover are also designed as cones opening towards the turbine wheel. Also this structural method ensures the force amplification which is typical for conical clutches and the particularly rigid design of the ring piston.

A structurally suitable design proposes that a damper output part is connected rotationally secured with the turbine wheel in its radially outer area to support on the output side the damper element whilst the support on the drive side is provided by a damper drive part connected rotationally secured with the ring piston.

This damper output part can preferably be a ring part welded to the turbine wheel and having entrainment fingers projecting in the direction of the friction disc of the ring piston.

The damper output part is however preferably designed as a leaf spring , connected rotationally secured to the ring piston and provided with arms projecting on the side of the clutch friction disc pointing towards the turbine wheel of the torque converter and enclosing the damper spring elements as well as with entrainment members supported circumferentially on one end.

Details of the control method according to the invention for use in vehicles with internal combustion engine as drive and a torque transfer system with a torque converter and a lock-up clutch parallel thereto will now be explained with reference to the accompanying drawings together with the advantages which can be achieved by this control method compared to the known control processes, wherein a lock-up clutch is shown by way of example.

In the drawings:

Fig. 1a is a diagrammatic view of a torque transfer system with a torque converter and a friction clutch mounted parallel thereto and bridging the converter;

Fig. 1b is a view of the engine torque as a function of the engine speed;

Fig. 1c shows the losses at the converter in comparison;

Fig. 1d shows the output loss as a function of the vehicle speed;

Fig. 1e shows the effect of the converter design on the pulling force;

Fig. 2 is a semi-sectional view of the torque transfer system corresponding to the diagrammatic illustration in Fig. 1a with a converter and a lock-up clutch as well as with a diagram of the associated pressurised medium control;

Fig. 3 is a diagrammatic view of the division of the engine torque into a torque to be transferred by the torque converter and a torque to be transferred by the lock-up clutch in dependence on the slip which occurs at the converter and the friction clutch bridging same;

Fig. 4 shows the engine speed and differential speed at the converter in dependence on the time when accelerating a motor vehicle with a gear change process with converter lock-up controlled in dependence on torque according to the invention;

Fig. 5 shows corresponding to Fig. 4 the output torque over the time when accelerating the vehicle with a gear change process with torque-controlled converter lock-up;

Fig. 6 is a view as in Fig. 4 of the speed behaviour when accelerating and with slip-controlled converter lock-up;

Fig. 7 corresponding to Fig. 6 is a view as in Fig. 5 of the output torque over the time during acceleration with slip-controlled converter lock-up;

Fig. 8 is a view as in Figs. 4 and 6 of the speed

behaviour when accelerating with converter lock-up opened during a gear-change process and closed again after the gear change;

Fig. 9 corresponding to Fig. 8 is a view as in Figs. 5 and 6 of the output torque over the time during accelerating with converter lock-up opened during a gear change process and closed again after the gear change;

Fig. 10 is a diagrammatic illustration showing the path of the pressure difference acting on the lock-up clutch in dependence on time to predetermine the value of the pressure difference which is desired after a scanning interval;

Fig. 11a shows the torque transfer system with a friction clutch bridging a hydrodynamic converter;

Figs. 11b and 11c show a temperature distribution in converter lock-up clutches;

Figs. 11d and 11e show the maximum surface pressure in converter lock-up clutches;

Fig. 12 is a diagrammatic view of the division of the engine torque into a torque to be transferred by the torque converter and a torque to be transferred by the lock-up clutch in dependence on the slip which occurs at the converter and the friction clutch bridging same;

Fig. 13 shows in a primary characteristic field of a "hard" converter the pump torque over the pump speed with the speed ratio turbine/pump as parameter;

Fig. 14 shows in a secondary field the turbine torque of the "hard" converter over the turbine speed;

Fig. 15 shows the output characteristic field of a converter of conventional "hard" design;

Fig. 16 is a view as in Fig. 13 of the primary characteristic field of a "soft" converter with the pump torque over the pump speed and the speed ratio turbine/pump as the parameter;

Fig. 17 shows in a secondary characteristic field of the "soft" converter according to Fig. 16 the turbine torque over the turbine speed;

Fig. 18 shows with the superimposed secondary characteristic fields according to Figs. 16 and 17 the conversion area which can be additionally used with the "soft" design of the converter;

Fig. 19 is a view as in Fig. 15 of the output characteristic field of the converter designed soft according to Fig. 18;

Fig. 20 shows the effect of the converter design on the losses;

Fig. 21 shows rotary vibrations of the engine and gearbox;

Fig. 22 shows a load change behaviour with conventional spring damper;

Fig. 23 shows the effect of the slip on the vibration behaviour;

Fig. 24 shows the load change behaviour with and without slip;

Fig. 25 shows the required slip without and with mini-torsion damper;

Fig. 26 is a "bubble chart" for the interaction of the slip, mini-torsion damper, cone design and adaptive control;

Fig. 27 shows with various diagrams examples of criteria for designing the k_{me} factor and the k_{me} characteristic field;

Fig. 28 is a flow diagram of a method for the torque control with adaption of a converter lock-up clutch;

Fig. 29 shows various diagrams for explaining the additive

and multiplicative correction factors which can be used for the torque control of a clutch;

Fig. 30 is a flow diagram of a torque control for a clutch;

Fig. 31 is a torque transfer system with a friction clutch bridging a hydrodynamic converter;

Figs. 32 and 33 show details of the torsion vibration damper according to Figure 24;

Fig. 34 shows a possible torsion characteristic line for the damper of a lock-up clutch;

Fig. 35 is a view of the output characteristic field of a "soft" converter;

Fig. 36 shows the sum frequency of the slip;

Fig. 37 shows the losses when driving uphill;

Fig. 38 is a diagrammatic illustration of the control method;

Fig. 39 is a diagram of the pulling force.

The torque transfer system 10 shown in Figs. 1a and 2 comprises a torque converter 11 and a lock-up clutch 12 which is connected in parallel with the torque converter and can be operated by pressurised flow medium. The torque transfer system is in active connection through a shaft 13 only indicated) with an internal combustion engine (not shown) and in turn is in driving connection on the output side through an output shaft 14 with an automatic gearbox which is mounted at the output in the drive train and is likewise not shown.

As shown by the diagrammatic semi-sectional view of the torque transfer system 10 in Figure 2 in conjunction with the pressure control diagram the torque converter 11 is a conventional flow converter. This flow converter consists of a converter cover 16 connected with the output of an

internal combustion engine, a pump wheel 17 forming together with the converter cover the converter housing, a turbine wheel 18 connected in turn by an output shaft with the automatic gearbox (not shown), as well as of a guide wheel 19 mounted between the pump and turbine wheel. The friction clutch 12 bridging the converter is mounted between the turbine wheel 18 and converter cover 16 and has a clutch disc 20 which is connected rotationally secured with the turbine wheel of the converter wherein the friction lining 21 of the clutch disc 20 interacts with a counter face 22 of the converter cover 16. The friction clutch furthermore has a rear chamber 24 facing the turbine wheel 18 and a front chamber 25 facing the converter cover 16.

The converter is a flow gearbox with pump 17, turbine 18 and guide wheel 19. Without slip it does not transfer any torque. With a constant input speed the following applies: the higher the slip the higher the torque. Figure 1b shows this connection for a rigidly braked output wherein the solid line in Figure 1b represents a conventional converter and the broken line represents a soft converter.

A converter is called soft when it has a higher slip with the same torque, that is also when it transfers less torque with the same slip. The softer converter thus opposes the engine with less resistance. If the driver requires a higher torque then the softer converter builds up higher speed differences.

The higher speed differences cause the so-called rubber band effect, the vehicle reacts delayed to the accelerator, ie it does not depend on the accelerator (petrol).

However: Most emission tests begin with a cold phase. If the engine reaches slightly higher speeds in this phase then

it becomes warmer more quickly and the emissions are clearly better.

With a given engine speed the soft converter opposes the engine with a lower torque. If the vehicle is at engine idling speed, then the engine must overcome the converter torque, and the losses thereby become less when the vehicle is stationary in Figure 1c with a soft converter. In Figure 1c for example with the vehicle stationary the loss of a soft converter is shown at 0.95 wherein the loss for a conventional converter is given at 1.6.

When an output torque is demanded, eg at a given vehicle speed at a given climb, the slip becomes greater with a soft converter, such as shown in Figure 1b, and thus the losses are also greater (Figure 1d).

As opposed to the clutch a torque converter can increase the torque. This torque conversion is higher with the same diameter and with a soft converter.

If the conversion is increased with the same diameter, then the converter becomes softer. The higher conversion leads to the pulling force (and thus the acceleration capacity) rising (Figure 1e).

The converter 11 is in known way provided with pressurised flow medium from a pressurised medium source (not shown in further detail) through a pipe 30 opening on the pump wheel side into the converter housing, wherein the pressure control is provided by a control valve 31 which is controlled in turn by a control element 32. The pressurised flow medium is discharged through a pipe (not shown) to a condenser 33 which is only indicated. In addition to biasing the turbine wheel 18 the pressure of the pressurised

flow medium acts on the outflow side of the pump wheel 17 in the rear chamber 24 of the friction clutch 12, biases the clutch disc 20 and presses this against the counter face 22 of the converter cover 16 interacting with the friction lining 21. Since according to the invention the clutch is driven with slip in all operating areas, the gap between the friction lining 21 of the clutch disc 20 and the counter face 22 of the converter cover 16 interacting therewith becomes greater or smaller in dependence on the slip and produces a throttled pressurised flow medium biasing of the front chamber 25 which extends between the clutch disc 20 and the converter cover 16. The pressurised flow medium biasing of the front chamber 25 can be controlled by means of a valve connected to this chamber through a pipe 34 so that an adjustable differential pressure acting between the rear and front chambers determines the torque which can be transferred by the friction clutch 12.

In view of the parallel arrangement of the converter 11 and the friction clutch 12 bridging same the engine torque equals the sum of the torques transferred by the converter and pump wheel and by the clutch , thus
$M_{Motor} = M_{Kupplung} + M_{Pumpenrad}$.

The gear torque, if one disregards the losses in the transmission system, is equal to the sum of the torques transferred by the converter and turbine wheel, thus
$M_{Getriebe} = M_{Kupplung} + M_{Turbinerad}$ or $M_{Kupplung} + (M_{Pumpenrad} \times \text{conversion})$.

Dividing the engine torque into a torque which is to be transferred by the converter and a torque to be transferred by the lock-up friction clutch is shown in Figure 3 in dependence on slip. It is evident that as the slip increases so the proportion of the engine torque transferred by the converter rises and correspondingly the torque

transferred by the clutch drops.

With the control method according to the invention, the slip is not controlled but in dependence on the operating state of the engine the proportion of the engine torque to be transferred by the friction clutch is determined and the differential pressure on the friction clutch required for transferring the predetermined torque is set by a computer unit, such as a micro processor. The slip is then produced by itself.

Figure 4 shows over the time the engine speed 40 and the differential speed 41 on the converter during accelerating and when changing up gear, such as from second to third gear. As a result of the acceleration the engine speed rises in second gear at first until it triggers the gear change process and then drops during the gear change process which begins at 42. The differential speed at the converter however remains constant at first, then rises sharply however during the gear change process. After changing from second to third gear the engine speed and differential speed at the converter drop at 43, namely the latter after a slight over swing to a value remaining constant at a higher level than before the gear change process. This is shown in Figure 4. The engine speed however rises slightly again in view of the preconditioned acceleration in third gear. It can be seen that at no point in time does the friction clutch bridging the converter stick. Rather driving is carried out with slip in all operating areas.

Of particular interest is the output torque 44 shown in Figure 5 over the time corresponding to Figure 4, wherein this output torque drops sharply at the beginning of the gear change process, then rises steeply during the phase of great slip as a result of the torque increase caused

thereby, and drops back again to a value corresponding to third gear at the end of the gear change process without any noticeable after vibrations 46 which moreover fade away immediately.

Figures 6 and 7 show that with the slip-controlled converter bridging the conditions are quite different during the gear-change process. Figures 6 and 7 also relate to changing from second to third gear in an accelerating vehicle.

As shown in Figure 6, the engine speed 40' rises in second gear until triggering the gear-change process at 42' whilst the differential speed 41' at the converter and thus the ensuing slip remains constant. At the beginning of the gear-change process at 42' the engine speed drops whilst the differential speed at the converter rises. After changing into third gear the engine speed and differential speed at the converter drop down again.

Since with a slip-controlled converter lock-up the intention is to keep the differential speed at the converter constant even during the gear-change process the gear change takes longer than with a torque-controlled converter lock-up because the turbine of the converter cannot yield. At the end of the gear change process sticking occurs on the lock-up friction clutch at 47 because the slip control can only act when a deviation has been set, and then only with a speed restricted by the setting elements and regulator stability. Finally as shown in Figure 6 the slip 41' is only set back to the level existing before the gear change process after the gear change process which has taken more time.

Even with a slip-controlled converter lock-up, the output torque 44' drops sharply at the beginning of the gear change

process in order to then rise steeply in the same way as with a torque-controlled converter lock-up and at the end of the actual gear change process to drop back to a value corresponding to third gear with noticeable after vibrations 46' which only gradually fade away.

It is apparent that with a slip controlled converter lock-up, the speed gradient and speed difference at the end of the gear change are very great. This is the reason why at the end of the gear change process the friction clutch sticks and in view of the then completely bridged converter the said after vibrations arise in the drive train.

Figures 8 and 9 also show similar to Figures 4 and 5 the acceleration of a vehicle with a gear change process wherein the converter lock-up is opened during the gear change process and closed after changing into a higher gear.

From Figure 8 it can be seen that until a gear process is triggered at 42'' the engine speed 40'' rises, whilst the differential speed 41'' at the converter drops slightly. During the actual gear change process the engine speed then drops corresponding to a change into a higher gear. The speed difference 41'' at the converter rises when introducing a gear change process in order to then drop again at the end of a gear change and to pass to zero at the end of a predetermined time as a result of the closing of the converter bridging at 48. With the driven torque the conditions are at first quite similar to those with the torque control of the converter lock-up according to the invention but lead to the quickly fading over vibrations 46'' directly at the end of the gear change process with a drop in the differential speed to zero, thus with the complete closing of the friction clutch bridging the converter it leads to considerable gear change jolts with

only slowly fading vibrations 49 in the drive train.

As shown by the comparison of the control concept according to the invention with reference to Figures 4 and 5 with the slip-controlled converter bridging according to Figures 6 and 7 and the control concept with the converter bridging according to Figures 8 and 9 opened during the gear change process and closed after changing gear, in the case of the torque-controlled converter bridging according to the invention substantially lighter gear change jolts occur than with other control concepts. This is due to the fact that during the gear change the converter bridging driven any way with predetermined slip yields and the differential speed can rise accordingly.

In the circuit diagram according to Figure 10 the curve 50 shows the path of the differential pressure ΔP acting at the lock-up clutch in dependence on the time. Starting from the initial differential pressure Δp_{Start} the pressure difference rises over the time sharply at first, which is indicated by the tangent 51 adjoining Δp_{Start}, in order then to weaken gradually in its rise and finally to approach asymptotically an ideal differential pressure indicated by chain-dotted line 52. This takes place through a stepped approach in that starting from a differential pressure Δp_n at a time t_n the differential pressure Δp_{n+1} is determined according to the equation given in patent claim 29 after a scanning interval Δt at a time t_{n+1}, the gradient of the pressure difference required after the time interval Δt is calculated and this gradient is set by the hydraulic system and finally the sequence is repeated continuously until the ideal value of the pressure difference indicated by the chain-dotted line 52 is reached.

The torque transfer system 60 shown by way of example in Figure 11a is a hydrodynamic torque converter 61 with a lock-up clutch 62 and a damper unit 63 acting between the torque converter and lock-up clutch.

The torque converter 61 comprises a pump wheel 65 which is in rotationally secured drive connection with an internal combustion engine (not shown), a turbine wheel 67 in active connection with a hub 66 on the output side, a guide wheel 68 mounted fixed in the flow circuit between the pump wheel and turbine wheel, and a converter cover 70 connected rotationally secured with the pump wheel and enclosing the turbine wheel.

The converter cover 70 is connected rotationally secured with the pump wheel 65 and provides the drive connection thereof with the internal combustion engine through entrainment studs 71,72 projecting on the side remote from the pump wheel and holding a flywheel (not shown) of the internal combustion engine.

Between the turbine wheel 67 and converter cover 70 is a ring piston 74 set central with the axis of rotation of the converter and comprising a shaped sheet metal part. This ring piston is set radially inwards with a sealing hub 75 on a counter sealing hub 76 extending away from the hub part 66 connected rotationally secured with the turbine wheel, and radially outwards is designed as a clutch friction disc 78 with a conical friction face 79.

The conical friction face 79 of the friction disc 78 of the ring piston 74 equipped with a suitable lining interacts with a correspondingly conically designed counter friction face 80 of the converter cover 70 which is connected rotationally secured with the pump wheel 65. The cones of

the interacting friction faces open according to the design provided with shading lines set further out towards the side pointing away from the turbine wheel 65. In view of this design a gusset-like ring space enclosed radially outwards by the converter cover 70 is formed between the peripheral area of the turbine wheel 67 and the conically designed clutch friction disc 78 of the ring piston.

The advantages of the cone design 79 are produced in particular through the stiffer structure and the greater friction surface:

* The loss of power which can be eliminated is clearly greater, and with the same loss of power the maximum oil temperature is lower. This helps to solve problem 5 (control speed), problem 2 (pulling of the engine) and the rest of problem 3 (control parameter problem).

* The strain on the lining is reduced through the more even surface pressure.

* The transferable torque becomes higher. Many single disc lock-up clutches are nowadays right at the limit of their performance capacity. The transferable torque drops further through the flow effects through the cooling oil stream.

* The weight and mass inertia torque is lower since the sheet metal chosen can be thinner owing to the stiffer construction.

* The converter distention becomes smaller and the controllability is thus improved.

The maximum temperature locally occurring plays an important role in the service life of the oil. The temperature is

kept low by a lining cooling, see Figures 11b, 11c.

The supporting surface becomes smaller through the lining cooling. This is however no problem with the cone since the lining pressure is more even (Figs. 11d and 11e). Moreover the transferable torque drops through the oil stream; the amplification effect of the cone 79 helps here.

The damper unit 63 is housed in the gusset-like ring space with ring-like damper spring elements 82 which are supported circumferentially on one side on damper drive parts 83 connected rotationally secured with the ring piston 74 and with their other ends on damper output parts 84 connected rotationally secured with the turbine wheel 67.

The damper drive parts 83 are designed like leaf springs, are mounted on the side of the ring piston 74 facing the turbine wheel 67 and are connected rotationally secured by means of rivets 85 to the piston in the area between the ring piston sealing hub 75 and the clutch friction disc 78. On the side remote from the friction face 79 of the clutch friction disc 78 there are arms 86,87 engaging round the damper spring elements 82 and extending from the damper drive parts 83 following the contour path of the ring piston 74, and also entrainment members 88,89 supporting each spring element on an end side.

The damper output parts 84 are ring segments welded to the peripheral area of the turbine wheel 67 whereby entrainment fingers 990 project from the ring segments towards the clutch friction disc 78 of the ring piston 74 and provide the support of the damper spring elements 82 at the other ends. The spring elements are thus housed between the entrainment members 88, 89 of the damper drive parts 83 and the protruding entrainment fingers 90 of the damper output

parts 84.

The damper unit 43 of the converter 41 is preferably designed on the main driving area which is shown in Figures 18 and 19 in the form of shaded faces. A damper designed as this kind which is shown in view of the complete converter bridging which is taken into consideration in this main driving area, ensures a substantially better damping of the rotary vibrations than would be possible if the damper was designed for a larger driving area. Furthermore a particularly compact converter construction is produced.

The lock-up clutch shown by way of example in the drawing and explained above has a pressure chamber 92 on the front side between the ring piston 74 and the turbine wheel 67 and also has a rear pressure chamber 93 between the ring piston and converter cover 70. The clutch friction disc 78 is operated into its clutch position interacting with the counter friction face 80 of the converter cover 70 as a result of biasing the pressure chamber 92 on the front side with pressurised flow medium and setting the torque to be transferred by the friction clutch takes place in dependence on the differential pressure acting between the pressure chamber 93 on the front side.

The input torque introduced through a flywheel (not shown) which is connected rotationally secured to the converter cover by the entrainment studs 71,72 projecting from the converter cover 70 to the side remote from the torque converter acts directly on the pump wheel 65 when the lock-up clutch 62 is open and is then transferred to the output hub 66 through the turbine wheel 67 owing to the hydraulic medium flow caused thereby.

When the lock-up clutch is completely closed however and the

friction disc 787 of the ring piston 74 interacts slip-free with the counter friction face 80 of the converter cover 70 then a direct mechanical transfer of the input torque to the turbine wheel 67 takes place through the damper spring elements 82 and then from the turbine wheel through the drive hub 66 fixedly connected to same to an output train in active connection with the automatic gearbox connected in on the output side.

If in dependence on a differential pressure acting between the front and rear pressure chambers 92,93 of the lock-up clutch the lock-up clutch operates with slip, then the input torque introduced through the converter cover 70 is divided up in dependence on slip into a torque transferred by the lock-up clutch 62 on one side and a torque transferred by the converter 61 on the other side, as shown diagrammatically in Figure 12.

The torque transfer from the lock-up clutch 62 to the turbine wheel 67 and the hub 66 on the output side connected rotationally secured to the wheel guarantees an effective compensation of the irregularities in the torque introduced. Owing to the arrangement of the damper spring elements 82 in the peripheral area between the friction disc 78 of the ring piston 74 and of the turbine wheel 67 it is guaranteed that comparatively large spring paths prevail.

The friction clutch 12, 14 can be controlled according to the invention so that it is at least partially at least at times closed in all the forward gears. In other words a slip control of the clutch is provided both in the first gear and from the first gear whereby a complete closing can also take place.

The conical friction faces of the converter cover 70 and of

the friction disc 78 can however also be designed as cones inclined towards the turbine wheel, as shown in the design with the closely shaded lines and at 70a and 78a. The damper springs 82 can then be stored radially further inwards, eg above the hub 66.

With the conventionally designed torque transfer systems the lock-up clutch which is completely open in the lower gears, is switched on in the upper gears. In the interest of a good overall degree of efficiency and to restrict the build up of heat the converters are made "hard". Figure 13 shows the primary characteristic field of a "hard" converter with the pump torque through the pump speed and the speed ratio turbine/pump as the parameters.

Figure 13 also shows a characteristic field of a drive motor with engine output torque over the engine speed agreeing with the turbine speed.

Finally Figure 13 also shows the main driving area shaded which comprises roughly the speed range between 750 and 2000 rpm.

The secondary characteristic field shown in Figure 14 shows the turbine torque over the turbine speed indicating the efficiencies in the various performance areas for the hard converter according to the characteristic field according to Figure 15.

The output characteristic field shown in Figure 15 in which the turbine torque of the converter is recorded over the turbine speed, shows the conversion area in which as the speed rises the turbine torque sharply drops, as well as the coupling area adjoining the conversion area. Furthermore the main driving area shown as a closely shaded surface is

entered in the output characteristic field.

With conventional torque transfer systems with a "hard" converter design in the interest of a good overall degree of efficiency with a restriction on the build up of heat, the torque increase drops sharply with rising speed. In the average speed area therefore only a slight torque increase takes place and in the upper speed area finally none at all.

In the primary characteristic field according to Figure 16 is shown the pump torque over the pump speed with the speed ratio turbine/pump as parameters of a "soft" converter. The characteristic lines of the "soft" converter have a substantially flatter path for the same parameters as in Figure 13. The conversion area extends over the average into the upper speed range.

As shown in Figures 17 this leads to a secondary field which is much wider compared to the secondary field of a hard converter shown in Figure 14. Thus with a soft converter design there are substantially greater acceleration reserves available which in many cases make changing down when accelerating unnecessary.

These acceleration reserves are shown in particular in Figure 18 in which the secondary field according to Figure 14 and belonging to a hard converter is placed over the secondary field according to Figure 17 and belonging to the soft converter. With the soft converter design the hatched area between the two solid lines of the two converters is obtained for a torque increase.

There is also shown in Figure 19 similar to Figure 15 the output characteristic field of a torque transfer system with soft converter design. The usable conversion area has

become larger compared to the output characteristic field in Figure 15 by the amount lying above the hatched line. Furthermore the main driving area shown as the closely shaded surface and the area with minimum slip are also entered in this characteristic field.

Furthermore operating points 1, 2 and 3 are entered in Figure 18. With the converter designs produced with "hard" and "soft" designs the following slips and degrees of efficiency could be detected in these operating points:

	"hard" converter		"soft" converter	
	slip s %	Efficiency η	slip s %	Efficiency η
Point 1	65	0.547	75	0.388
Point 2	40	0.789	60	0.669
Point 3	2	0.980	2	0.980

It is evident that in the lower and average speed areas with a "soft" converter design the degree of efficiency does indeed remain behind the degree of efficiency of a "hard" converter but noticeably increased slip occurs and thus improved torque increase. In operating point 3 in Figure 18 the slip and degrees of efficiency are however the same with a hard and soft converter design.

As a result of the dynamic behaviour of the hydraulic and mechanical systems, with a too rapid increase in the amount of a parameter affecting the division of the torque to be transferred by the torque transfer system between converter and friction clutch it can lead to the generation of vibrations of different frequency through too much jolt or sticking of the friction clutch.

In order to avoid such vibrations a suitable design of the invention proposes that the setting of a newly calculated amount, differing from the previous amount, of a parameter affecting the division of the torque to be transferred between the converter and friction clutch, preferably the differential pressure, takes place delayed after a function in dependence on time.

Similarly the setting of a newly calculated amount, differing from the previous amount, of a parameter affecting the division of the torque to be transferred between the converter and friction clutch is possible delayed after a function in dependence on the gradient of the engine speed.

The loss of performance with a given driving condition (weight, gradient) decreases with a rising conversion compared with a soft converter without higher conversion, but it is in general higher than one with a stiffer converter Figure 20. With a large slip the losses are despite a simultaneously softer converter no higher than with a stiffer converter because the higher torque conversion improves the degree of efficiency (Area A in Figure 20).

Without a lock-up clutch it is necessary to find a compromise between stationary losses and acceleration capacity (the converter should therefore rather be soft) and between losses in the driving operation (here the converter should rather be stiff). Here the limits are provided by the converter physics. These were brought up to strength in recent years. The converters thus designed are rather stiff.

The effect of the lost power can be restricted by a

conventional lock-up clutch with torsion damper and without slip. There are however limitations. For reasons of comfort (humming, rattling and load changes) these lock-up clutches can only be used in the upper gears and only at higher speeds. Nevertheless there are still certain comfort losses which remain.

To show the problems of humming and rattling Figure 21 shows the vibration amplitude above the engine speed - excited by an engine order. Depending on the engine excitation and humming sensitivity of the vehicle it may only be possible to bridge at a higher speed. As is known however the engine is operated for most of the time at relatively low speeds. Therefore the fuel savings are limited.

A further problem is the load change behaviour and the behaviour when switching the bridging on and off (Figure 22). If the driver provides throttle in the bridged state he at first receives a jerky vibration instead of the desired increase in pulling power. The lock-up then opens which in unfavourable cases can even lead initially to a short torque breakthrough. The driver only then receives the desired increase in pulling power. When closing the lock-up it can again lead to an excitation in the drive train.

Even when changing gear there may be problems with comfort which is why the bridging is usually opened prior to a gear change.

These effects are strongest in the lower gears and therefore with conventional lock-up systems bridging is only in 4th and 5th gear.

The power loss which occurs in first gear when driving

uphill is therefore not reduced by bridging. This loss of power also restricts the permissible softness of the converter with the given cooling output.

The "rubber band effect" cannot be prevented in many areas and therefore also does not allow the choice of a softer converter.

The converter lock-up system of the inventive idea consists of a slipping lock-up clutch in the cone design with mini-torsion dampers, an adaptive control and a soft converter.

It is a system with slip. What are the advantages which slip has over complete lock-up?

* Humming is reduced.
* The load change behaviour is improved.
* The gear change quality is significantly better (completely bridged not acceptable).
* The switch-on quality of the bridge is better (switch-on here means that when the converter is open the lock-up clutch is operated, but not absolutely to zero slip).

By reducing the humming excitation the bridging can be used earlier than with conventional systems (see Figure 23).

The load change behaviour and the preparation of the pulling power is significantly improved compared to conventional systems (see Figure 24). If the driver provides throttle there is no jerky vibration because the bridging slips through. Through this slipping the slip builds up and thus the converter torque.

Also no torque breakthrough occurs. The torque rises

continuously, through the increasing conversion beyond the engine torque. It can be bridged earlier, also in low gears and at low speeds.

If it provides so many advantages why then are slipping lock-ups still not used everywhere today? There are naturally also some possible problems in the case of slip (see Figure 25 for explanation):

1. At low speed the slip required to avoid humming is mostly relatively large, thereby the power loss is also large. If the slip is reduced then sticking occurs temporarily which will in many cases cause humming (Δn_1).
2. Many engines must not be pressed too much at high loads. If pressing of the engine at high loads is not permitted there are two possibilities. The engine speed is increased by completely opening the bridging or letting the bridging slip more strongly. If it is allowed to slip more strongly then higher power losses occur at the bridging (Δn_4).
3. To control slight slip is difficult. With "sharp" regulating parameters there are frequently regulating problems, one relaxes the parameters, the slip can clearly deviate. In many cases a control has advantages but a fluctuation in the slip is scarcely avoidable even here. It can lead to sticking (danger of humming) or to a lot of slip (higher losses).
4. The control is not accurate in any way. The lower the torque to be set then the more difficult it becomes to achieve an exact regulation or control.
5. The control is not fast in any way. With non-stationary processes the control path requires a setting time. In these phases the slip deviates. In order to avoid humming a certain slip must thus be observed. Again higher slip values are thereby

produced (Δn_2, Δn_3).

6. Losses occur at the bridging. A significant problem with slipping clutches is that regarding the service life. Mostly they bear the power losses for a long time. However after several thousand kilometres problems of pulling may start to appear. These pulling problems mostly have their cause in damage to the oil - the friction lining is generally still in order. The additives are damaged by local overheating and in time this has an effect on the entire oil. Even with a slight loss of power the lining must be cooled very well. The security against local overheating should be the greatest possible! To the loss of power through slip for removing vibrations and through points 2 and 3 comes still the loss of power when switching the bridging in and out. The lower the speeds and the higher the loads at which the bridging is switched on, the greater the loss of power - particularly if one is paying attention to a comfortable bridging switch.

These problems are overcome by a very simple torsion damper (which can also be designed only for partial load), the cone design and an adaptive control. The interaction of these system components is sown in a "bubble chart" (Figure 26). The bubbles with the thick outline represent the customer requirements and the more thinly outlined bubbles represent the components of the system according to the invention. The mini torsion damper: Advantages of this very simple torsion damper (see also Figure 25):

* Problem 1 (humming) can thus be solved. The impulses which occur with a temporary sticking are thus filtered and humming does not occur.
* Problem 3 (control parameter problem) can thus be partially solved. Also here the temporary sticking no longer has a negative effect.

* Problem 4 (control accuracy with low torque) can thus be solved. With low engine torque the bridging can be closed with a higher level torque since the torsion damper undertakes to remove vibrations.

The slip can be selected lower. In the resonance range of the damper the slip prevents excitation. Therefore no friction elements are necessary in the damper. The mini-torsion damper is lighter and cheaper than a conventional torsion damper.

The strongly interlinked connections shown in the bubble chart in Figure 26 will now be explained with reference to key words:

-> Soft converter => lower idling losses
but => greater losses in driving mode
Aim: to lower consumption => to bridge in all gears
\+ to increase conversion
=> Improvement in driving performances
Aim: to avoid humming => control inaccuracy
=> on average high slip necessary

-> Mini torsion damper + adaptive control
=> average slip can be reduced
\+ load change, gear change, L. and switch-on and response behaviour can be optimized
but non-stationary and with low engine speeds
=> great slip necessary
=> discharge heat
=> lower local oil temperature

-> Conical lock-up => large surface
=> good lining grooving possible
=> high cooling oil stream achievable

but -> transferable torque is lowered
=> reinforcement effect of the cone
=> torque transfer guaranteed

=> stiff form
=> even pressure
=> ideal strain on lining
=> constant friction value
=> good local temperature distribution

=> increase in service life

=> lower mass inertia torque
=> further improvement in driving performance

Diagrams 1 to 5 in Figure 27 show examples for criteria for designing the k_{me} factor or the k_{me} characteristic field which can be stored for example in a central processor unit (CPU) wherein + indicates good quality and - poor quality.

The k_{me} factor is shown on the abscissa of this diagram and the tendency effect of the criteria in relation to the size of the k_{me} factor is shown on the ordinate axis. As is apparent from a comparison between the idealized characteristic lines entered in the diagrams, the different criteria are in part contradictory, ie they are opposite, considered over the k_{me} factor. For this reason when considering several of these criteria it is necessary to evaluate their priority or importance according to the case of use and the desired vehicle behaviour. As can be seen from diagram 1, the acoustics or noise behaviour cannot be improved in any way by selecting a very small k_{me} factor because otherwise as a result of the high slip in the lock-up clutch an inadmissibly high thermal load can arise on

same or on the converter. The negative influence of too much slip in the lock-up clutch can be seen from the diagram 2 of Figure 27. There are thus boundary conditions which must not be understepped or overstepped. Between the still acceptable boundary conditions which should not be understepped or overstepped it is however possible to change the k_{me} factor. As already mentioned, the k_{me} factor can be varied in dependence on the ensuing operating conditions wherein this variation can be carried out stepwise or continuously between certain boundary values. The k_{me} factor can advantageously be varied in dependence on the condition levels of the vehicle. These condition levels of the vehicle or drive can be detected by a processor so that the k_{me} factor allocated to these condition levels can be set or detected. This k_{me} factor can be read off from a stored characteristic field for example.

In many operating conditions of an internal combustion engine it can be advantageous if the lock-up clutch is biased so that it can transfer the complete net torque just arising at that time and released by the combustion engine. A biasing of the lock-up clutch in this way can be advantageous particularly in the lower operating range of an internal combustion engine wherein it is then particularly expedient if the lock-up clutch has a damper which is designed for this partial load area. A damper of this kind thus has a lock-up or stop torque which is less than the maximum or nominal torque released by the combustion engine. This stop torque can be in the order of between 30 and 60% of the nominal torque of the internal combustion engine. The effect of such a damper can be seen from diagram 1 in Figure 27. The use of such a vibration damper makes it possible to solve at least in part the acoustic problems which arise in the lower operating area of an internal combustion engine in conjunction with a comparatively large

k_{me} factor.

Figure 28 is a block circuit diagram or flow diagram of a torque control with adaption which will be explained in further detail below. The operation of the converter lock-up clutch can thereby be provided by an electro-hydraulic setting member.

According to Figure 28 at 1 first the drive torque of the drive assembly such as in particular an internal combustion engine is calculated from various input values. The values used here comprise at least two of the following, namely speed of the drive assembly, load lever position or throttle position of the fuel supply, underpressure in the inlet manifold system, injection time, consumption etc. At 2 linking 1 takes place which causes a correction of the drive torque. This correction is carried out by means of correction factors which are supplied by the system adaption characterised at 12. These correction factors can compensate the deviations from the desired state which occur in the system , namely by compensating these deviations through additive, multiplicative and/or non-linear proportions.

At 3 the correct k_{me} factor for each operating state is fixed or detected. This factor represents the torque ratio $M_{Kupplung}$ to $M_{Antrieb\ korrigiert}$ which is to be set by the control, as a value previously fixed for each operating point in the manner of a characteristic field from the relevant selected evaluation of the criteria indicated in Figure 27. The design of the damper possibly present of the lock-up clutch is thereby of particular importance since with the presence of such the k_{me} factor can be kept constant at least over a comparatively large section of the operating area of the internal combustion engine or hydrodynamic torque converter.

At 4 the calculation of the ideal clutch torque is carried out by means of the relevant k_{me} factor and the corrected drive torque of the drive assembly. At 5 a further correction of the ideal clutch torque can take place by the additive, multiplicative and/or non-linear proportions resulting from the system adaption 12. The link 2 can thus be provided. For many cases it is sufficient if only one of the two links 1,2 is present wherein the link 1 should preferably be maintained.

At 6 the calculation takes place of the setting value from the corrected ideal clutch torque and the inverse transfer function of the path which the lock-up clutch represents. At 7 the regulator output value can be calculated on the base of the setting value detected at 6 and of the inverse transfer function of the setting member. The setting member can advantageously be formed by an electrohydraulic setting member. Advantageously a proportional valve can be used or even a pulse-width modulated valve. At 8 a back coupling of the setting values can take place in the form of a regulation or adaption. This back coupling can however also be dispensed with. At 9 a measurement of the actual clutch torque can take place, eg through a torque sensor or expansion measuring strip (DMS). Instead of the measurement of the actual clutch torque taking place at 9 it is also possible to calculate this torque from the condition levels and from the vehicle and converter physics. To this end the engine characteristic field and/or the converter characteristic field or values of these characteristic fields can be stored in a processor or in a central processor unit. Furthermore a characteristic field representing the torque transfer capacity of the converter lock-up clutch or values representing same can be recorded for this purpose.

If a detection of the actual clutch torque takes place both according to point 9 and point 10 then a balancing of the measured actual clutch torque with the actual clutch torque calculated from the model can take place. The balancing can thereby take place as a logical link minimum-maximum formation or as a plausibility comparison.

In the system adaption characterised by 12 in Figure 28 the following comparisons inter alia can take place and the corresponding corrections can thus be carried out.

a) Comparison of corrected ideal clutch torque and actual clutch torque wherein this comparison can also take place long term, eg by observing deviations through a co-rotating time window.

Comparison of corrected drive torque and recalculated drive torque wherein this comparison can also take place long term eg by observing deviations over a co-rotating time window.

Evaluation of extra signals such as eg switching additional assemblies on and off, such as eg air conditioning, compressors etc, gear change.

b) Detection of the system deviations detected under a) in additive, multiplicative and/or non-linear proportions of $M_{Antrieb}$ and $M_{Kupplung}$ and division resulting therefrom into the corresponding adaption loops 1 and 2 or into links 1 and 2.

The detection of the corresponding proportions of $M_{Antrieb}$ and/or $M_{Kupplung}$ can take place for example according to the three diagrams of Figure 29.

In the diagram according to Example 1 of Figure 29 an actual torque path and ideal torque path are recorded over the time wherein over the time the actual torque has a sudden change. This sudden change can be due for example to the switching on of an extra assembly such as a compressor. The change, caused by this extra assembly, in the actual torque which is available for the converter can be taken into account through an additive proportion by means of which the engine torque can be corrected accordingly.

An actual torque path and an ideal torque path over the time are likewise shown in the diagram according to Example 2 of Figure 29. From this diagram it can be seen that the ratio between the torques allocated to a certain time remains substantially the same although the difference between the torques changes. Deviations of this kind between the actual torque and ideal torque can take place by a multiplicative proportion. A path of this kind between the actual torque and ideal torque can be due for example to the friction value path or friction engagement existing between the friction faces of the lock-up clutch. This concerns a multiplicative clutch proportion.

An actual torque path and an ideal torque path over the time are likewise shown in the diagram according to Example 3 of Figure 29. As can be seen, the two torques change over the time wherein however over the time the difference between the two torques is at least approximately constant. Such deviation between the actual torque and the desired ideal torque can be compensated by an additive proportion. Such a path between the actual torque and the ideal torque can be due to a deviation in the setting size for the lock-up clutch.

Figure 30 shows a flow diagram of a torque control with a very simple adaption. The control of the lock-up clutch takes place electrohydraulically through a proportional valve or a pulse-width modulated valve. The output signal of the control computer or control output valve is a setting current which adjusts proportionally to a scanning ratio adjoining eg the pulse-width modulated output of the computer. The clutch torque results from the pressure difference controlled in this way at the converter lock-up clutch or between the two pressure chambers of the lock-up clutch. The system adaption is restricted to the adaptive correction of the drive torque whose deviation results from the difference between the ideal and actual clutch torque.

With an embodiment according to Figure 30 the link 2 and the return of the correct drive torque (M-an-korr) are omitted compared to Figure 28.

In Figure 30, the DP-ideal is detected at 6, namely as a function of the ideal clutch torque as the main value and if necessary still in dependence on the corrected drive torque (M-an-korr) and turbine speed (n-turbine) as parameter.

The function block 7 according to Figure 28 is in Figure 30 divided into two sub-function blocks, namely in 7a and 7b. The sub-function blocks 7a and 7b are each allocated a return coupling 8a and 8b. Input value of the inverse transfer function of the setting member (7=7a and 7b) is the ideal pressure difference (DP-soll) calculated in block 6. The output value is formed by the associated scanning ratio as control output value. The adjoining setting member is divided into the electric setting member proportion which is formed by an end phase and the valve development, as well as in the hydraulic setting member proportion which is decisive for the corresponding pressure biasing of the converter

lock-up clutch. The input value of the electric setting member proportion is the scanning ratio. This is converted on the output side into an actual current. In dependence on this actual current (I-Ist) the hydraulic setting member proportion sets a corresponding pressure biasing of the converter lock-up clutch. This takes place by setting a corresponding pressure difference between the chambers, eg 24, 25 according to Figure 2 of the converter lock-up clutch. The block 7a represents the inverse function of the hydraulic setting member proportion in which the associated ideal current (I-soll) is calculated from the ideal pressure (DP-soll). This part of the setting member has a return coupling of the measured actual pressure (DP-ist) in the form of a pressure adaption which is represented by the block 8a. This pressure adaption 8a supplies the corrected ideal current (I-soll-korr). The second part 7b of the inverse transfer function 7 of the setting member represents the electric proportion which calculates the associated scanning ratio from the corrected ideal current. The input value I-Soll-R for the inverse transfer behaviour of the electric setting member proportion is thereby calculated from the control deviation I-Soll korr= =-I-Ist (I-Ist measured according to the valve winding) with a PID regulator.

The numbering 1 to 12 of the individual blocks selected in Figure 30 corresponds substantially to the numbering of the individual blocks of Figure 28. In this way the individual function blocks of the special electrohydraulic design according to Figure 30 can be related to those of the general design according to Figure 28.

The individual terms contained in Figure 30 have the following meaning:

DP-Soll= Ideal pressure difference at the lock-up or converter lock-up clutch. Corresponds to the pressure difference between the pressures prevailing in the chambers existing either side of the piston.

DP-Ist= Actual pressure difference between the two chambers of the converter lock-up clutch.

p-nach= Pressure after lock-up or converter lock-up clutch, thus pressure in the chamber 25 or in the return line 34 according to Figure 2.

I-Soll= Ideal current for the electrohydraulic valve.

Delta-n Speed difference between pump wheel and turbine wheel, thus delta-n= n-pump wheel - n-turbine wheel.

Values marked with "*-korr" correspond to values corrected by adaption.

The condition values of the vehicle indicated in Figure 30 in front of the block marked 10 contain the slip in the lock-up clutch or in the converter.

As can be seen from Figure 30 the speed difference delta-n = n-pump wheel - n-turbine wheel represents no control value, as is the case with the known slip controls. With the torque control according to the invention this speed difference delta-n is used as the condition level of the path to be controlled for observing possible torque deviations which then in the adaption through corresponding links acts back with correction on the control. The observed torque values can be stored eg in the manner of a

co-rotating time window over a certain time lapse in order to detect the proportions of the deviations at the clutch and engine. This takes place in the system adaption marked by 12.

The control according to the invention furthermore has the advantage that the adaption of the disruptive proportions of the drive torque can take place even when the lock up or converter lock-up clutch is fully opened, thus with $k_{me} = 0$. To this end the nominal drive torque (M-an) is compared with the torque adjoining the converter, which takes place in the link 1 according to Figure 28 and with the method step 2 of Figures 28 and 30. Through this adaption any possible deviations of the drive torque (M-an) in the open state of the lock-up clutch can already be considered in anticipation of a later closing of the lock-up clutch. To this end the torque adjoining the converter is detected in the system adaption 12, namely the converter characteristic field is preferably recorded and stored in this system adaption. Thus by detecting the speed difference between the turbine wheel and pump wheel it is possible to detect the ensuing torque. This converter torque is then compared with the nominal drive torque (M-an) of the motor or drive assembly. This drive torque (M-an) can be derived from a stationary engine characteristic field recorded in the block 1 according to Figures 28 and 30, namely as a result of the measured condition levels, such as in particular the engine speed, load lever position, consumption, injection amount or injection time etc. The speed difference between the turbine wheel and pump wheel can be detected in block 10.

Furthermore it is possible to detect the converter torque already in block 10 wherein the converter characteristic field is then recorded in block 10.

The torque transfer system 110 shown as an embodiment in Figure 31 is a hydrodynamic torque converter 111 with lock-up clutch 112 and a damper unit 135 acting between the torque converter and lock-up clutch.

The torque converter 111 comprises a pump wheel 117 in rotationally secured driving connection with an internal combustion engine (not shown), a turbine wheel 118 actively connected with a hub 114 on the output side, a guide wheel 119 mounted in the flow circuit between the pump wheel and turbine wheel, and a converter cover 116 enclosing the turbine wheel and connected rotationally secured with the pump wheel.

The converter cover 116 is connected rotationally secured with the pump wheel 117 and produces its drive connection with the internal combustion engine through entrainment areas 116 projecting on the side remote from the pump wheel 118 whereby a drive disc (not shown) of the combustion engine can be fixed on these entrainment areas.

Between the turbine wheel 118 and the radial area of the converter cover 116 is a ring piston 136 set central with the axis of rotation of the converter and comprising a shaped sheet metal part. This ring piston is set radially inwards on an output hub 114 which is connected rotationally fixed to the turbine wheel 118 and forms radially outwards a conical area which is fitted with a suitable lining 121. The ring piston 136 interacts with a correspondingly conical counter friction face 122 of the converter cover 116.

The lock-up clutch 112 has a rear pressure chamber 124 between the ring piston 136 and the turbine wheel 118 and a front pressure chamber 125 between the ring piston 136 and the converter cover 116. The piston 136 is operated in its

clutch position interacting with the counter friction face 122 by biasing of the front pressure chamber 125 with flow medium. The size of the torque to be transferred by the friction clutch 112 is in dependence on the differential pressure set between the pressure chambers 124, 125.

The torsion damper 135 is designed so that its lock-up torque or stop torque is less than the nominal torque, thus the maximum torque of the combustion engine driving the torque converter 110. This means that the energy accumulators 137 of the torsion damper 135 are designed so that they cannot resiliently absorb the entire torque of the internal combustion engine. The relative torsion between the input part 138 of the torsion damper 135 connected rotationally secured to the piston 136 and the flange like output part 139 can take place through blocking of the windings of the springs 137 or preferably through stops provided between the input part 138 and output part 139. The output part 139 of the damper 135 is connected rotationally secured with the turbine hub 114 in known way by axial push-in connection formed by gearings.

As can be seen from Figure 32, the input part 138 interacting with the energy accumulators 137 can be formed by segment shaped components 140 wherein each two such components 140 mounted back to back are provided diametrically opposite each other. The pairs of segment shaped components 140 are connected rotationally secured with the piston 136 by rivet connections 141. Figure 33 is a plan view of the flange-like output part 139. The flange 139 has a ring shaped foundation body 139a and two diametrically opposite radial extension arms 142 with recesses 143 for the energy accumulators 137. The extension arms 142 are set axially between the pairs of components 140. The segment shaped components adjoining each other

back to back in pairs - viewed circumferentially -form between their fastening areas 144 housing pockets 145 for the extension arms 142. In Figure 33 the stop contours 146 formed by the segment shaped components 140 for the extension arms 142 are interrupted by broken lines. The piston 136 has circumferentially spaced axial indentations which form projections 147 directed towards the turbine wheel 118 and supporting the fastening areas 144 of the segment shaped components 140 facing the piston 136. The components 140 likewise have recesses 148 for the springs 137. These recesses 148 are in the illustrated embodiment axially aligned with the recesses 143 of the output part 139. With the embodiment according to Figures 31 to 33 the energy accumulators 137 are set play free in the recesses 143 and 148. In many cases it can however also be expedient if at least one of the springs 137 has play relative to a recess 143 and/or 148. Also at least one of the springs 137 can be installed with a certain pretension in a window 143 and/or a window 148.

Since the torsion damper 135 is only designed for a partial load area this can be particularly simple in construction so that a cost effective production is possible.

The torsion damper 135 can be designed according to an embodiment according to the invention so that about 40 to 50% of the maximum, thus nominal torque of the internal combustion engine can be transferred through the springs 137. The relative angle of travel covered by the energy accumulators 137 between the input part 138 and output part 139 can as shown in Figure 34 lie in the order of 5°. In Figure 34 the relative angle of travel between the input part 138 and output part 139 of the damper 135 is shown during the pulling operation of the motor vehicle. In the pushing operation this relative angle of travel can be of

equal size or have another value. Also the turning stiffness of the torsion damper 135 can be quite different in the pulling direction and pushing direction. This is achieved by corresponding dimensions of the windows 143 and 148 and of the springs 137. Also the torsion damper 135 can have a multi-phase characteristic line wherein the characteristic line areas which correspond to the pushing operation and pulling operation can likewise have different paths.

From Figure 34 it can be seen that the torsion damper 135 is bridged at 5 deg. angle or passes to a stop and the torque transferable by the elasticity or compression of the springs 137 is restricted to about 45 Nm. A torsion damper 135 designed in this way can advantageously be used in connection with hydrodynamic torque converters which have a slip-controlled lock-up clutch. The stop torque of 45 Nm is suitable for engines which have a nominal torque in the order of 80 to 200 Nm.

The lock-up torque of the damper 135 is preferably measured so that this preferably covers the entire main driving area of a motor vehicle. As main driving area is to be considered that area which is used most frequently taken over the entire operating time of the motor vehicle. This main driving area preferably comprises at least the areas of the engine characteristic field which relate to the FTP75 cycle and/or for the ECE cycle (town, 90 kmh, 120 km/h). The main driving area is thus that area in which the vehicle is mostly driven. As a result of the traffic infrastructures existing in individual countries, this driving range can be somewhat different between the individual countries.

In the output characteristic field of a torque converter 111

shown in Figure 35 with soft converter design the main driving area is shown as a closely shaded surface. Furthermore the conversion range of the torque converter is shown in Figure 35. In this conversion range the lock-up clutch 112 is open. The main driving range is enclosed by an area in which driving is preferably with a minimum slip in the lock-up clutch 112. The main driving area extends from a lower speed A to an upper speed B. The lower speed A thereby corresponds at least substantially to the idling speed which can lie in the order of 700 to 800 revolutions. The upper speed limit B can lie in a speed area between 2000 and 3000 revolutions and for example have the value 2200 rpm. The area with slip can have an upper speed limit C which can correspond to the maximum speed of the internal combustion engine, advantageously however can also lie below same and can have for example a value between 3000 and 4000 rpm.

Through the design of the torsion damper 135 according to the invention the torque converter 111 can be completely bridged in the main driving area, thus the lock-up clutch 112 can be operated without slip, ie the k_{me} factor is greater than 1, preferably 1.1. In this main driving area the vibration isolation between the internal combustion engine and the gearbox connected to the output side takes place practically completely through the torsion vibration damper 135. Only peak torques are taken up by the slip in the lock-up clutch 112. To this end the lock-up clutch 112 is controlled and regulated in the main driving area so that it transfers a comparatively low torque in relation to the maximum torque of the engine although this torque is greater than the torque of the engine just arising .

In the area with slip the lock-up clutch 112 is controlled and regulated so that a certain slip is provided between the

friction faces 121, 122 of the lock-up clutch 112. As a result of this slip a relative rotation exists between the pump wheel 117 and turbine wheel 118.

In the area with slip (k_{me} factor is less than 1, eg 0.9) according to Figure 35 the disruptive torque irregularities still occurring here are mainly damped by slip.

In the main driving area and in the area with slip for better vibration isolation insofar as conditions occur in the driving train with high vibration amplitude, such as for example with resonance, load change impact or the like the transferable torque of the lock-up clutch 112 can be reduced. This can take place by changing the k_{me} factor.

As can be seen from Figure 34 the torsion damper of the lock-up clutch 112 can also be designed so that this has adjoining an angle of travel with relatively low turning angle stiffness a comparatively small angle of travel in which the turning stiffness amounts to a multiple of that of the first angle. In Figure 34 this second angle of travel extends over 2°. The turning stiffness in this second angle of travel can amount to 7 to 15 times the turning stiffness in the first angle of travel. With the embodiment shown in Figure 34 the turning stiffness in the first angle of travel is in the order of 8 Nm/° and in the second angle of travel in the order of 70 Nm/°.

In the main driving area according to Figure 35 the torque transferable by the lock-up clutch 112 is set by the k_{me} factor to about 1.1 to 1.2 times the engine torque actually arising. The control or regulation of the torque transferable by the lock-up clutch 112 can take place in the main driving area so that the torque transferable by the

lock-up clutch 112 does not drop below a minimum value. This value should amount to at least 1% of the nominal torque of the engine. The minimum torque transferable by the lock-up clutch 112 in the main driving area can amount for example to 5 Nm. This lower limit can however be moved down or up according to the type of use. Thus the minimum torque transferable in the main driving area by the lock-up clutch 112 can also be set to a value which is very close to, and preferably slightly lower than, the maximum engine torque in the main driving area.

In the area designated "area with slip" in Figure 35 the torque transferable by the lock-up clutch 112 is set through the k_{me} factor to 0.8 to 0.95 times the torque momentarily arising of the internal combustion engine. The torque transfer capacity of the lock-up clutch 112 is thus dependent on the relevant ensuing torque of the internal combustion engine which has to be transferred. In other words this means that with a rising torque of the combustion engine the torque transferable by the lock-up clutch also increases and with a reduction in the torque released by the combustion engine the torque transfer capacity of the lock-up clutch 112 likewise decreases.

Through the design of the converter lock-up clutch according to the invention and its control an optimum operation of a motor vehicle is possible from the energy points of view. Since driving is carried out with slip-free lock-up clutch in the operating conditions mainly used it is possible to achieve a considerable fuel saving compared to the converter lock-up clutches which are not bridged in these operating conditions or which operate with slip. The main speed range thereby lies between about 600 and 2200 to 3000 revolutions per minute or the mean value about 1800 rpm. In the main driving area thus the lock-up clutch is substantially closed

so that the prevailing engine torque is transferred by the lock-up clutch without significant slip. The vibration damping is carried out in the main driving area by the rotary vibration damper 135 provided in the force or torque flow of the converter lock-up clutch 112. The torsion damper 135 is thereby provided with a comparatively small angle of travel and the stop torque of the torsion damper corresponds roughly to the upper boundary torque of the main driving area. This upper boundary torque can amount to 15 to 50% of the maximum engine torque depending on the motorization and vehicle weight. With a damper constructed in this way vibrations can prevail in the driving area with lower drive torques to produce a disturbing humming. Disturbing load change reactions in the drive train are suppressed or avoided by the comparatively small angle of travel of the torsion damper. The load change impacts are restricted in that on exceeding the stop torque or lock-up torque of the damper the friction faces of the lock-up clutch slip relative to each other. The torque to be transferred is thereby restricted. The torque peaks are dampened by slip in the lock-up clutch. Above the main driving area or in the driving area in which the ensuing torque are greater than the boundary torque transferable by the damper the lock-up clutch is controlled so that a slip is present. Disruptive load change reactions are avoided by the slip thus set. In speed areas or in torque areas above the main driving area in which there are no disruptive vibration excitations the clutch can likewise be closed to a torque value which is greater than the ensuing engine torque. For certain speed ranges in which disruptive excitations are present the lock-up clutch can again be opened to slip. The latter can be expedient particularly with the onset of resonance speed.

Also in the main driving area or in the area of

comparatively small engine torques it can be expedient when passing through resonances to open the lock-up clutch and to reduce considerably the torque transferable by same. Through the design and control or regulation of the lock-up clutch according to the invention so-called humming noises are to be avoided which cannot be removed through a partially closed, thus slipping lock-up clutch, namely as a result of the sticking/sliding states which occur between the friction faces of this lock-up clutch.

Supplementary explanations to understand the invention or inventions as well as additional advantages of the designs and constructions according to the invention compared to the prior art are apparent from the following description.

Which control strategy was selected for the system of the converter lock-up clutch according to the invention? The problems which appear with slip regulation were already discussed above. The basic problem lies in that first a control deviation must take place before the regulator reacts. Furthermore there are areas in which the ideal suggestion cannot be reached, eg no higher slip can be regulated than would occur with an open converter. With gear changes there is a negative effect if the regulator operates against the course of the gear change. If for example in the case of high speed gear changes the slip is kept too low then at the end of the gear change it can result in sticking and thus to loss of comfort. It is possible to think up solutions to all these control problems -however usually this does not represent the optimum solution. The LuK control concept therefore works torque-controlled, and the system deviations are compensated by adaption. The lock-up torque is determined from the engine torque:

$M_{Uberbruckung} = M_{Motor}$ * Bridging factor

This means that no ideal slip is set. This is also shown in the sum frequency diagram in Figure 36 wherein the solid line evaluates the prior art and the broken line represents the idea according to the invention.

Whether the bridging is completely opened or closed with slip is initially fixed from the energy points of view. An example:

With extreme uphill driving (3600 kg, 12%) at low speed the bridging cannot be completely closed since eg the pulling force reserves are not sufficient or the engine must not be pressed too severely. Then it is constantly compared whether the overall losses are lower if bridging with slip is provided or if the bridging is completely opened, cf Figure 37.

If the driver wishes to increase the pulling force he raises the load lever position. At first the engine torque rises. If this torque is not sufficient the driver raises the load lever position further in order to signify his wish for additional acceleration. With conventional systems it is mostly switching back to increase the pulling force through a shorter translation. With the WL system it is first checked whether by opening the lock-up clutch an increase in pulling force can be expected. This is then the case if the converter were located in the conversion area after opening. If this is so then the lock-up clutch is opened, otherwise it switches back. This checking takes place constantly. In order to improve the interaction it is advisable to adapt the gearbox switching lines to this concept. This conformity is particularly effective in combination with a soft converter (see next chapter). This philosophy can be

shown in approximation in a circuit line diagram (Figure 38).

Even through these measures (slip, cone, mini damper and adaptive control) a clear improvement in the consumption is achieved since in all gears bridging is possible. The converter causes an even clearer improvement.

At the beginning of this article reference was made to the design of the converter. Since a bridging is not possible in all areas with the systems which are customary nowadays the converter must be made correspondingly stiff. With the WL concept it is possible to utilize the advantage of a soft converter and to avoid the disadvantages. The advantages are basically a better pulling power and lower stationary losses. The disadvantages - in many areas under load higher losses and the "rubber band effect" - are avoided through the lock-up clutch which can be used all the time.

With this design further significant advantages were achieved. The driving performance is clearly increased and the consumption noticeably reduced. Furthermore the emissions are improved beyond proportion. The test cycles begin with a cold phase. When the bridge is opened the engine reaches its operating temperature substantially quicker with a soft converter which has a favourable effect on the emissions.

In the acceleration from 0 to 100 km/h there are no great differences between conventional 4-speed gearboxes and 5-speed gearboxes since the translations are practically the same in the first gears. With the 4-speed gearbox with the LuK WL system (through the further converter design) there are clear acceleration advantages over the conventional 5-speed gearbox. Also in fuel consumption clear improvements

can be reached both in comparison with the conventional 4-speed gearbox and in comparison with the conventional 5-speed gearbox. Considerable improvements in the emissions can also be expected.

A cost-effect solution with many advantages: the combination of the converter lock-up clutch system with a 4-speed gearbox.

The system of the converter lock-up clutch in combination with a 4-speed gearbox can, in relation to driving performance and consumption, produce similar advantages to a 5-speed gearbox with conventional lock-up clutch and this with overall clearly lower weight and costs than in the case of a 5-speed gearbox (quite apart from the possible reductions in the case of development costs).

The combination of a soft converter with widely expanded 4-gears shows in the pulling force diagram in most areas even a higher pulling force than a 5-speed gearbox with conventional converter (Figure 39). It is also possible to see that in areas with lower load the 5-speed gearbox must change over two gear stages and not the 4-speed gearbox with the converter lock-up system, ie the switching frequency of the gearbox is reduced. The greater gear jumps are taken up by the softer converter.

The invention is not restricted to the embodiments described and illustrated but comprises in particular variations which can be formed by a combination of features and elements described in connection with the present invention. Furthermore individual features or functioning methods described in connection with the figures can be taken alone to represent an independent invention.

The applicant reserves the right to claim as being essential

to the invention further features which up until now have only been disclosed in the description, more particularly in connection with the figures. The patent claims filed with the application are thus only proposed wordings without prejudice for achieving wider patent protection.

This application is one of a series of applications, all based on application 9503415.3

Application 9503415.3 describes and claims a method for controlling a torque transfer system which is in active connection with the output of a drive assembly, possibly an internal combustion engine, and is in driving connection through an output shaft with an automatic gearbox wherein the torque transfer system has a torque converter and a friction clutch mounted parallel thereto, a measured value detection system and a central computer unit wherein the force biasing of the friction clutch and thus the torque transferred by same can be consciously changed in conjunction with a central computer unit, characterised in that the torque to be transferred by the friction clutch is detected in dependence on the torque of the drive assembly and the force biasing of the friction clutch which is required to transfer the predetermined clutch torque is calculated and adapted accordingly wherein a minimum slip is independently set between the drive and output of the calculated clutch torque and deviations from the ideal state are compensated long-term by corrections.

Application 9803688.2 (Agents ref: P1508.P3B) describes and claims a method for controlling a torque transfer system which is in active connection with the output of a drive assembly, possibly an internal combustion engine, and is in driving connection through an output shaft with an automatic gearbox wherein the torque transfer system has a current

converter and a friction clutch mounted parallel thereto, a measured value detection system and a central computer unit wherein the force biasing of the friction clutch and thus the torque transferred by same can be consciously changed in conjunction with a central computer unit, characterised in that the torque to be transferred by the friction clutch is detected in dependence on the operating state of the drive assembly according to the torque equation

$$M_{Kupplung} = k_e \cdot K_{korr} \cdot M_{Antriebsaggregat}$$

with $k_e = k_{me}$ as the torque division factor and
k_{korr} as the correction factor

and the force biasing of the friction clutch which is required to transfer the predetermined clutch torque is calculated and adjusted wherein the slip is independently set between the drive and output of torque transfer system in dependence on the size of the torque division factor k_e which is constant over the entire operating range of the drive train and the correction factor k_{korr} compensates any deviations of each special drive train from the ideal state.

Application 98 03676.7 (Agents ref: P1508.P3D) describes and claims a torque transfer system for the drive train of a vehicle equipped with a gear-change gearbox, more particularly a motor vehicle with internal combustion engine drive, with a torque converter which is in driving connection with a drive assembly of the vehicle and is in active connection through an output shaft with an automatic gearbox connected on the output side, with a friction clutch mounted parallel to the hydrodynamic converter and designed to be operated by the pressurised flow medium and having a pressure chamber mounted between a turbine wheel of the converter and a ring piston actively connected to friction

disc on one side, and another pressure chamber between the ring piston and a converter cover on the other side, which pressure chambers are designed so that a differential pressure existing between these pressure chambers determines the torque transferable by the friction clutch, with a measured value detection system, a central computer unit and with a hydraulic system which in conjunction with the computer unit produces a deliberate change in the differential pressure between the two pressure chambers and thus in the torque transferable by the friction clutch, wherein the friction clutch is controlled in all driving gears and that the torque converter has a torque conversion > 2.5.

Application 98 03681.7 (Agents ref: P1508.P3E) describes and claims a drive system with internal combustion engine and slip-controlled lock-up clutch for a hydrodynamic torque converter wherein the lock-up clutch contains a torsion damper whose stop torque is less than the nominal torque of the internal combustion engine.

Application 98 03683.3 (Agents ref: P1508.P3F) describes and claims a method for controlling a lock-up clutch for a hydrodynamic torque converter which (clutch) is slip-controlled in dependence on the torque to be transferred wherein a control orientated from the energy and performance-related points of view is active at least in all forward gear stages of a gearbox

Application 98 03685.8 (Agents ref: P1508.P3G) describes and claims a drive system with internal combustion engine characterised in that at least with acceleration processes a device determines whether by opening the lock-up clutch in the same gear an increase in tensile force can be achieved

through torque conversion and in this case opens, otherwise the gearbox is shifted back at least one gear stage.

Application 9803687.4 (Agents ref: P1508.P3H) describes and claims a drive system with internal combustion engine characterised in that at least with acceleration processes a device determines whether by opening the lock-up clutch with the same translation an increase in tensile force can be achieved through torque conversion and in this case opens, otherwise the gearbox changes or increases the translation.

PATENT CLAIMS

1. Lock-up clutch for a hydrodynamic torque converter of a torque transfer system, with a pump wheel, turbine wheel, guide wheel and a converter cover which is central with the axis of rotation, connected rotationally secured with the pump wheel and encloses the turbine wheel, wherein a central ring piston mounted between the converter cover and turbine wheel is designed radially outwards as a conical clutch friction disc and has radially inwards a sealing hub which is set on a counter sealing hub which is connected rotationally secured with the turbine wheel.

2. Lock-up clutch as claimed in Claim 1, wherein the clutch friction disc of the ring piston and the counter friction face of the converter cover interacting with same are designed as cones opening towards the side pointing away from the turbine wheel.

3. Lock-up clutch as claimed in Claim 1 or Claim 2, wherein at least one ring-shaped damper element of a damping unit is set circumferentially between a damper drive part connected rotationally secured with the ring piston, and a damper output part connected rotationally secured with the turbine wheel, and is mounted between the radially outer area of the turbine wheel and the clutch friction disc of the ring piston having a friction surface.

4. Lock-up clutch as claimed in any preceding claim, wherein the damper output part is connected rotationally secured with the turbine wheel in the radially outer area thereof.

5. Lock-up clutch as claimed in any preceding claim, wherein the damper output part is a ring part welded to the

turbine wheel and having entrainment fingers protruding in the direction of the clutch friction disc of the ring piston.

6. Lock-up clutch as claimed in any preceding claim, wherein the damper output part is designed as a leaf spring and is connected rotationally secured with the ring piston and has on the side remote from the friction surface arms protruding from the clutch friction disc and engaging round the damper spring elements and also has on one end circumferentially supporting entrainment members.

7. Lock-up clutch as claimed in any preceding claim, wherein the clutch is for use with an automatic gearbox which is a stepped gearbox or a continuous or stepless gearbox.

8. Method for controlling a torque transfer system which includes a lock-up clutch as claimed in any preceding claim, wherein the torque transfer system is in active connection with the output of a drive assembly, and is in driving connection through an output shaft with an automatic gearbox wherein the torque transfer system has a torque converter and a friction clutch mounted parallel thereto, a measured value detection system and a central computer unit wherein the force biasing of the friction clutch and thus the torque transferred by same can be consciously changed in conjunction with a central computer unit, wherein the torque to be transferred by the friction clutch is detected in dependence on the torque of the drive assembly and the force biasing of the friction clutch which is required to transfer the predetermined clutch torque is calculated and adapted accordingly wherein a minimum slip is independently set between the drive and output of the calculated clutch torque and deviations from the ideal state are compensated long-

term by corrections.

9. Method as claimed in Claim 8, wherein the torque to be transferred by the friction clutch is detected in dependence on the torque of the drive assembly as claimed in the torque equation

$$M_{Kupplung} = k_{me} \cdot k_{korr} \cdot (M_{Antriebsaggregat} + M_{korr\_MOT}) + M_{korr\_WÜ}$$

with

$M_{Kupplung}$	-	Torque on the friction clutch
k_{me}	-	Torque division factor
k_{korr}	-	Correction factor for compensating multiplicative faults
M_{korr_MOT}	-	Correction torque for compensating faults adding to the engine torque

wherein a minimum slip is independently set between the drive and output of the torque transfer system in dependence on the size of the torque division factor k_{me} which is constant over the entire operating range of the drive train and any deviations from the ideal state are compensated long-term by the correction factor k_{korr} and the correction torques M_{korr_MOT} and $M_{korr\_WÜ}$.

10. Method as claimed in Claim 9, wherein the torque division factor k_{me} is a value dependent on the output speed.

11. Method as claimed in Claim 9, wherein the torque division factor k_{me} is a value dependent solely on the speed of the drive assembly.

12. Method as claimed in Claim 9, wherein the torque division factor k_{me} is a value dependent on both the speed and the torque of the drive assembly.

13. Method as claimed in Claim 8, wherein the torque division factor k_{me} is a value dependent on both the output speed and the torque of the drive assembly.

14. Method as claimed in any one of Claims 8 to 13, wherein the friction clutch can be operated by a pressurised flow medium and is designed so that two separate pressure chambers are formed between the friction clutch and the converter cover and between the friction clutch and the remaining converter housing and that a differential pressure existing between these pressure chambers determines the torque transferred by the friction clutch.

15. Method as claimed in any one of Claims 8 to 14, wherein with a torque transfer system with an internal combustion engine as the drive assembly its operating state is determined in dependence on the engine speed and on the throttle flap angle, in dependence on the engine speed and on the fuel through-put, in dependence on the engine speed and on the suction pipe underpressure or in dependence on the engine speed and the injection time.

16. Method as claimed in any one of Claims 8 to 15, wherein setting a newly calculated amount differing from the previous amount of a parameter which affects the division of the torque to be transferred between the converter and friction clutch, preferably the amount of differential pressure, is carried out delayed after a function in dependence on time.

17. Method as claimed in any one of Claims 8 to 16, wherein setting a newly calculated amount differing from the previous amount of a parameter which affects the division of the torque to be transferred between the converter and friction clutch, preferably the amount of differential

pressure, is carried out delayed after a function in dependence on the differential speed between the drive and output of the torque transfer system.

18. Method as claimed in any one of Claims 8 to 17, wherein setting a newly calculated amount differing from the previous amount of a parameter which affects the division of the torque to be transferred between the converter and friction clutch, preferably the amount of differential pressure, is carried out delayed after a function in dependence on the gradient of the engine speed.

19. Method as claimed in any one of Claims 8 to 18, wherein the differential pressure desired at the friction clutch is regulated by means of a PI or PID regulator wherein the control path from the differential pressure on the friction clutch required to determine a specific torque to be transferred by the friction clutch, to the adjusting differential pressure cannot be described clearly analytically.

20. Method as claimed in any one of Claims 8 to 19, wherein the desired differential pressure at the friction clutch is set in that a pressure-proportional signal such as a valve stream is derived from a characteristic line and set wherein any deviations which occur between the ideal and actual pressure are compensated by means of an I-return and the control path from the differential pressure at the friction clutch which is required to determine a specific torque which is to be transferred by the friction clutch, to the setting differential pressure cannot be described clearly analytically.

21. Method as claimed in any one of Claims 8 to 20, wherein the desired differential pressure is set at the friction

clutch in that a signal proportional with the desired differential pressure, such as a current or touch ratio, is calculated and regulated by means of a PI, I or PID regulator wherein the control path from the differential pressure at the friction clutch required for obtaining a specific torque which is to be transferred by the friction clutch, to the setting differential pressure cannot be described clearly analytically.

22. Method as claimed in any one of Claims 8 to 21, wherein deviations of the torque actually transferred by the friction clutch, from the desired torque are ascertained by measuring the adjusting slip between the drive and output of the torque transfer system and comparing same with ideal values.

23. Method as claimed in any one of Claims 8 to 21, wherein deviations of the torque actually transferred by the friction clutch from the desired torque are ascertained in that the torque transferred by the torque converter is calculated from its characteristic and thus the actual torque division between the converter and friction clutch is checked.

24. Method as claimed in any one of Claims 9 to 23, wherein deviations which occur between the torque actually transferred by the friction clutch and the desired torque are due to

multiplicative errors

$(k_{korr} \neq 0,\ M_{korr\_MOT} = 0,\ M_{korr\_WÜ} = 0$

errors adding to the engine torque

$(k_{korr} = 0,\ M_{korr\_MOT} \neq 0,\ M_{korr\_WÜ} = 0)$

errors adding to the clutch torque
($k_{korr} \neq 0$, $M_{korr\_MOT} = 0$, $M_{korr\_WÜ} \neq 0$)

errors both multiplying and adding to the engine torque
($k_{korr} \neq 0$, $M_{korr\_MOT} \neq 0$, $M_{korr\_WÜ} = 0$)

errors both multiplying and adding to the clutch torque
($k_{korr} \neq 0$, $M_{korr\_MOT} = 0$, $M_{korr\_WÜ} \neq 0$)

or errors multiplying and adding to both the engine torque and the clutch torque
($k_{korr} \neq 0$, $M_{korr\_MOT} \neq 0$, $M_{korr\_WÜ} \neq 0$)

and that the compensation of such errors is carried out with a time constant of several seconds in order to achieve only an adaptive character of the control.

25. Method as claimed in any one of Claims 9 to 24, wherein when a desire to accelerate is signalled by the driver the slip in the torque transfer system is increased by reducing the k_{me} factor and the increase in torque offered by the converter can be used as additional torque reserves.

26. Method as claimed in any one of Claims 9 to 25, wherein the slip in the torque transfer system is determined in all gears by the friction clutch whereby the degree of efficiency of the output transmission through the converter recedes into the background and a converter interpretation is permitted with regards to a high stall-speed and broad conversion range.

27. Method as claimed in any one of Claims 9 to 26, wherein the slip in the torque transfer system is determined in all translations by the friction clutch whereby the degree of

efficiency of the output transmission recedes in the background and a converter interpretation is permitted with regards to a high stall speed and broad conversion range.

28. Method for controlling a torque transfer system which includes a lock-up clutch as claimed in any preceding claim, wherein the torque transfer is in active connection with the output of a drive assembly, possibly an internal combustion engine, and is in driving connection through an output shaft with an automatic gearbox wherein the torque transfer system has a current converter and a friction clutch mounted parallel thereto, a measured value detection system and a central computer unit wherein the force biasing of the friction clutch and thus the torque transferred by same can be consciously changed in conjunction with a central computer unit, wherein the torque to be transferred by the friction clutch is detected in dependence on the operating state of the drive assembly as claimed in the torque equation

$$M_{Kupplung} = k_e \cdot K_{korr} \cdot M_{Antriebsaggregat}$$

with $k_e = k_{me}$ as the torque division factor and k_{korr} as the correction factor

and the force biasing of the friction clutch which is required to transfer the predetermined clutch torque is calculated and adjusted wherein the slip is independently set between the drive and output of torque transfer system in dependence on the size of the torque division factor k_e which is constant over the entire operating range of the drive train and the correction factor k_{korr} compensates any deviations of each special drive train from the ideal state.

29. Method for controlling a torque transfer system which

includes a lock-up clutch as claimed in any preceding claim, wherein the torque transfer is in active connection with the output of a drive assembly, possibly an internal combustion engine, and is in driving connection through an output shaft with an automatic gearbox wherein the torque transfer system has a current converter and a friction clutch mounted parallel thereto, a measured value detection system and a central computer unit wherein the force biasing of the friction clutch and thus the torque transferred by same can be consciously changed in conjunction with a central computer unit, wherein the torque to be transferred by the friction clutch is detected in dependence on the operating state of the drive assembly as claimed in the torque equation

$$M_{Kupplung} = k_e \cdot K_{korr} \cdot M_{Antriebsaggregat}$$

with $k_e = k_{me}$ as the torque division factor and k_{korr} as the correction factor

and the force biasing of the friction clutch which is required to transfer the predetermined clutch torque is calculated and adjusted wherein the slip is independently set between the drive and output of the torque transfer system in dependence on the size of the torque division factor k_e which is independent of the engine characteristic field and the correction factor k_{korr} compensates any deviations of each special drive train from the ideal state.

30. Method for controlling a torque transfer system which includes a lock-up clutch as claimed in any preceding claim, wherein the torque transfer is in active connection with the output of a drive assembly, possibly an internal combustion engine, and is in driving connection through an output shaft with an automatic gearbox wherein the torque transfer system

has a current converter and a friction clutch mounted parallel thereto, a measured value detection system and a central computer unit wherein the force biasing of the friction clutch and thus the torque transferred by same can be consciously changed in conjunction with a central computer unit, wherein the torque to be transferred by the friction clutch is detected in dependence on the operating state of the drive assembly as claimed in the torque equation

$$M_{Kupplung} = k_e \cdot K_{korr} \cdot M_{Antriebsaggregat}$$

with $k_e = k_{me}$ as the torque division factor and k_{korr} as the correction factor

and the force biasing of the friction clutch which is required to transfer the predetermined clutch torque is calculated and adjusted wherein the slip is independently set between the drive and output of the torque transfer system in dependence on the size of the torque division factor k_e which is dependent on the speed of the drive assembly alone and the correction factor k_{korr} compensates any deviations of each special drive train from the ideal state.

31. Method for controlling a torque transfer system which includes a lock-up clutch as claimed in any preceding claim, wherein the torque transfer is in active connection with the output of a drive assembly, possibly an internal combustion engine, and is in driving connection through an output shaft with an automatic gearbox wherein the torque transfer system has a current converter and a friction clutch mounted parallel thereto, a measured value detection system and a central computer unit wherein the force biasing of the friction clutch and thus the torque transferred by same can be consciously changed in conjunction with a central

computer unit, wherein the torque to be transferred by the friction clutch is detected in dependence on the operating state of the drive assembly as claimed in the torque equation

$$M_{Kupplung} = k_e \cdot K_{korr} \cdot M_{Antriebsaggregat}$$

with $k_e = k_{me}$ as the torque division factor and k_{korr} as the correction factor

and the force biasing of the friction clutch which is required to transfer the predetermined clutch torque is calculated and adjusted wherein the slip is independently set between the drive and output of the torque transfer system in dependence on the size of the torque division factor k_e which is dependent on both the speed and the torque of the drive assembly and the correction factor k_{korr} compensates any deviations of each special drive train from the ideal state.

32. Method as claimed in any one of Claims 28 to 31, wherein the friction clutch can be operated by a pressurised flow medium and is designed so that two separate pressure chambers are formed between the friction clutch and converter cover and between the friction clutch and the remaining converter housing and that a differential pressure existing between these pressure chambers determines the torque transferred by the friction clutch.

33. Method as claimed in any one of Claims 28 to 32, wherein with a torque transfer system having an internal combustion engine as the drive assembly its operating state is determined in dependence on the engine speed and throttle valve angle.

34. Method as claimed in any one of Claims 28 to 33, wherein with a torque transfer system having an internal combustion engine as the drive assembly its operating state is determined in dependence on the engine speed and inlet manifold underpressure.

35. Method as claimed in any one of Claims 28 to 34, wherein with a torque transfer system having an internal combustion engine as the drive assembly its operating state is determined in dependence on the engine speed and injection time.

36. Method as claimed in any one of Claims 28 to 34, wherein a torque to be transferred by the friction clutch and detected in the central computer unit in dependence on a torque change in the drive train and which (torque) differs from the momentary torque is set by the following measures:

- predetermining the value of any parameter X determining the torque to be transferred by the friction clutch and which (value) is desired after a scanning interval at time point tn+1, after a function which excludes undesired incidents, such as eg sticking of the friction clutch,
- calculating the gradient ΔX which is required to obtain the desired value of the parameter X after a time interval Δt,
- setting the calculated gradient ΔX by means of the hydraulic system through proportional regulation wherein as parameter the differential pressure P between the pressure chambers of the clutch is predetermined as claimed in the equation

$$\Delta P_{n+1} = (1 - ß) \cdot \Delta P_{Soll} + ß \cdot \Delta P_n$$

with

$\beta = f\ (T_v,\ t)$ and

- repeating the above sequence until an ideal value X_{Soll} is reached.

37. Method as claimed in any one of Claims 28 to 36, wherein a new value of the torque transferred by the friction clutch and detected in the central computer unit in dependence on a torque change in the drive train and is set by

- calculating the gradient ΔX of any parameter X determining the torque to be transferred by the friction clutch after a function which excludes undesired incidents, such as eg temporary sticking of the friction clutch,
- setting the desired gradient ΔX by means of the hydraulic system wherein the gradient of the pressure difference ΔP between the pressure chambers of the clutch is calculated as the parameter as claimed in the equation

  $$\Delta\Delta P = C_1 \cdot (\Delta P_{Soll} - \Delta P_n)$$

  with

  C1 = proportionality factor, and
- repeating the above sequence until the required ideal value X_{Soll} is reached.

38. Method as claimed in any one of Claims 28 to 37, wherein in operating cases where a reduction of the input torque at the torque transfer system is to be expected, such as when changing down gear or when switching on additional assemblies, a possible temporary sticking of the friction clutch is counteracted by reducing the torque transferred by the friction clutch in that either the torque division factor k_e or the correction factor k_{korr} is reduced by a predetermined value and after a function in dependent on

time is raised again to an optimum value for vibration isolation and fuel economy.

39. Method as claimed in any one of Claims 28 to 38, wherein in operating cases where a reduction of the input torque at the torque transfer system is to be expected, such as when changing the translation or when switching on additional assemblies, a possible temporary sticking of the friction clutch is counteracted by reducing the torque transferred by the friction clutch in that either the torque division factor k_e or the correction factor k_{korr} is reduced by a predetermined value and after a function in dependent on time is raised again to an optimum value for vibration isolation and fuel economy.

40. Method as claimed in any one of Claims 28 to 39, wherein in operating cases where a reduction of the input torque at the torque transfer system is to be expected, such as when changing down gears or when switching on additional assemblies, a possible temporary sticking of the friction clutch is counteracted by reducing the torque transferred by the friction clutch in that either the torque division factor k_e or the correction factor k_{korr} is reduced by a predetermined value and after a function in dependent on time is raised again to an optimum value for vibration isolation and fuel economy.

41. Method as claimed in any one of Claims 28 to 40, wherein the correction factor k_{korr} compensates for any deviations of each special drive train from the ideal state in that preferably in a fixed quasi stationary operating area the setting slip is measured with a time stagger which excludes vibrations and the slip is compared with ideal slip values which guarantee optimum vibration isolation with the highest possible fuel economy, and the factor k_{korr} is

balanced in the event of any deviation between the ideal and actual slip.

42. Method as claimed in any one of Claims 28 to 41, wherein when a desire for acceleration is signalled by the driver, preferably documented by the change in speed of the throttle valve angle, the slip in the torque transfer system is increased by reducing one of the factors k_e or k_{korr} and the torque increase offered by the converter can be used as additional torque reserve.

43. Method as claimed in any one of Claims 28 to 42, wherein the slip in the torque transfer system in all gears is determined by the friction clutch whereby the degree of efficiency of the output transmission through the converter recedes in the background and a converter interpretation is made possible with regard to the widest possible conversion range.

44. Method as claimed in any one of Claims 28 to 43, wherein the slip in the torque transfer system is determined with all translations by the friction clutch whereby the degree of efficiency of the output transmission recedes in the background and a converter interpretation is made possible with regard to the widest possible conversion range.

45. Torque transfer system for the drive train of a vehicle equipped with a gear-change gearbox, a lock-up clutch as claimed in any one of Claims 1 to 7, and a torque converter which is in driving connection with a drive assembly of the vehicle and is in active connection through an output shaft with an automatic gearbox connected on the output side, with a friction clutch mounted parallel to the hydrodynamic converter and designed to be operated by the pressurised

flow medium and having a pressure chamber mounted between a turbine wheel of the converter and a ring piston actively connected to friction disc on one side, and another pressure chamber between the ring piston and a converter cover on the other side, which pressure chambers are designed so that a differential pressure existing between these pressure chambers determines the torque transferable by the friction clutch, with a measured value detection system, a central computer unit and with a hydraulic system which in conjunction with the computer unit produces a deliberate change in the differential pressure between the two pressure chambers and thus in the torque transferable by the friction clutch, wherein the friction clutch is controlled in all driving gears and that the torque converter has a torque conversion > 2.5.

46. Torque transfer system for the drive train of a vehicle equipped with a gear-change gearbox, a lock-up clutch as claimed in any one of Claims 1 to 7, and a torque converter which is in driving connection with a drive assembly of the vehicle and is in active connection through an output shaft with an automatic gearbox connected on the output side, with a friction clutch mounted parallel to the hydrodynamic converter and designed to be operated by the pressurised flow medium and having a pressure chamber mounted between a turbine wheel of the converter and a ring piston actively connected to a friction disc on one side, and another between the ring piston and a converter cover on the other side, which pressure chambers are designed so that a differential pressure existing between these pressure chambers determines the torque transferable by the friction clutch, with a measured value detection system, a central computer unit and with a hydraulic system which in conjunction with the computer unit produces a deliberate change in the differential pressure between the two pressure

chambers and thus in the torque transferable by the friction clutch, wherein the friction clutch is controlled with all translations and that the torque converter has a torque conversion > 2.5.

47. Torque transfer system as claimed in Claim 45 or Claim 46, wherein the heat building up during the driving operation is calculated up by the computer unit and the actual heat balance thus obtained is compared with the amount of heat which is permissible as claimed in the structural design.

48. Torque transfer system as claimed in any one of Claims 45 to 47, wherein with extreme driving situations the slip is changed via the lock-up control and thus the amount of heat building up is reduced.

49. Torque transfer system as claimed in any one of Claims 45 to 48, wherein with the exception of the extreme situations, such as starting up, accelerating, climbing uphill, the lock-up is always operated with very small slip.

50. Torque transfer system as claimed in any one of Claims 45 to 49, wherein a damper unit acting between the turbine of the converter and the friction disc of the lock-up clutch is designed for the partial load area.

51. Torque transfer system for the drive train of a vehicle equipped with a gear-change gearbox, a lock-up clutch as claimed in any one of Claims 1 to 7, and a torque converter which is in driving connection with a drive assembly of the vehicle and is in active connection through an output shaft with an automatic gearbox connected on the output side, with a friction clutch mounted parallel to the hydrodynamic converter and designed to be operated by the pressurised

flow medium and having a pressure chamber mounted between a turbine wheel of the converter and a ring piston actively connected to a friction disc on one side, and another between the ring piston and a converter cover on the other side, which pressure chambers are designed so that a differential pressure existing between these pressure chambers determines the torque transferable by the friction clutch, with a measured value detection system, a central computer unit and with a hydraulic system which in conjunction with the computer unit produces a deliberate change in the differential pressure between the two pressure chambers and thus in the torque transferable by the friction clutch, wherein the friction clutch is controlled so that in all forward gears a partial closing takes place at least at times.

52. Torque transfer system for the drive train of a vehicle equipped with a gear-change gearbox, more particularly a motor vehicle with internal combustion engine drive, with a torque converter which is in driving connection with a drive assembly of the vehicle and is in active connection through an output shaft with an automatic gearbox connected on the output side, with a friction clutch mounted parallel to the hydrodynamic converter and designed to be operated by the pressurised flow medium and having a pressure chamber mounted between a turbine wheel of the converter and a ring piston actively connected to a friction disc on one side, and another between the ring piston and a converter cover on the other side, which pressure chambers are designed so that a differential pressure existing between these pressure chambers determines the torque transferable by the friction clutch, with a measured value detection system, a central computer unit and with a hydraulic system which in conjunction with the computer unit produces a deliberate change in the differential pressure between the two pressure

chambers and thus in the torque transferable by the friction clutch, wherein the friction clutch is controlled so that in all forward translations a partial closing takes place at least at times.

53. Drive system with internal combustion engine and slip-controlled lock-up clutch as claimed in any one of Claims 1 to 7 for a hydrodynamic torque converter, wherein the lock-up clutch contains a torsion damper whose stop torque is less than the nominal torque of the internal combustion engine.

54. Drive system as claimed in Claim 53, wherein the stop torque amounts to between 10 and 60% of the maximum torque of the internal combustion engine, preferably between 25 and 50%.

55. Drive system as claimed in Claim 53 or Claim 54, wherein the damper has no inherent friction device.

56. Drive system as claimed in any one of Claims 53 to 55, wherein the damper allows a relatively small angle of travel in the order of ± 2 to 6°, preferably ± 3 to 8°.

57. Drive system as claimed in any one of Claims 53 to 56, wherein the damper has a stiffness of 7 to 30 Nm/°.

58. Method for controlling a lock-up clutch as claimed in any one of Claims 1 to 7, which clutch is slip-controlled in dependence on the torque to be transferred wherein a control orientated from the energy and performance-related points of view is active at least in all forward gear stages of a gearbox.

59. Method for controlling a drive system with internal

combustion engine as claimed in any one of Claims 53 to 56 and 58, wherein the torque control of the lock-up clutch is divided at least into two areas of which the first extends in the area of 10 to 60 %, preferably from 15 to 50% of the maximum torque of the internal combustion engine and the second lies above this.

60. Method as claimed in Claim 57, wherein in the first area the torque transferable by the lock-up clutch is more than the torque of the combustion engine arising at the time.

61. Method as claimed in Claim 60, wherein the torque transferable by the lock-up clutch amounts to 1.0 to at least 1.2 times the relevant ensuing torque of the internal combustion engine.

62. Drive system as claimed in any one of Claims 53 to 57, wherein in the first area in conditions with high vibration amplitude in the drive train, that is for example in the case of resonance, load change impact or the like, the transferable torque of the lock-up clutch can be reduced.

63. Drive system as claimed in any one of Claims 53 to 57 and 62, wherein the stop torque of the torsion damper corresponds at least approximately to the torque of the internal combustion engine occurring at the end of the first area.

64. Drive system as claimed in any one of Claims 53 to 57 and 62 and 63, wherein at least over a partial area of the first area the minimum torque transferable by the lock-up clutch is kept greater than 1% of the nominal torque of the internal combustion engine.

65. Drive system as claimed in any one of Claims 53 to 57 and 62 to 64, wherein at least over a partial area of the first area the torque transferable by the lock-up clutch is kept at least approximately at a constant value.

66. Drive system as claimed in any one of Claims 53 to 57 and 62 to 65, wherein at least the major proportion of the characteristic field of the internal combustion engine used in the main driving area (for example, the areas of the engine characteristic field which are relevant for the FTP75-cycle and/or for the ECE-cycle [town 90 km/h, 120 km/h] falls within the first area.

67. Drive system as claimed in any one of Claims 53 to 57 and 62 to 66, wherein the first area extends from idling speed up to a maximum 3000 rpm, preferably up to a maximum of between 2000 and 2500 rpm.

68. Drive system as claimed in any one of Claims 53 to 57 and 62 to 67, wherein in the second area the torque transferable by the lock-up clutch amounts to 0.6 to < 1.0 times the relevant ensuing torque of the internal combustion engine, preferably 0.8 to 0.9 times.

69. Drive system as claimed in any one of Claims 53 to 57 and 62 to 68, wherein a device determines whether by opening the lock-up clutch in the same gear an increase in tensile force can be achieved through torque conversion and in this case opens, otherwise the gearbox is shifted back at least one gear stage.

70. Lock-up clutch for a hydrodynamic torque converter substantially as herein described with reference to any one embodiment described herein.

71. Lock-up clutch for a hydrodynamic torque converter substantially as herein described with reference to any one embodiment shown in the accompanying drawings.

The Patent Office

120

Application No:	GB 9803672.6	**Examiner:**	Tom Sutherland
Claims searched:	1 - 7	**Date of search:**	16 April 1998

Patents Act 1977
Search Report under Section 17

Databases searched:

UK Patent Office collections, including GB, EP, WO & US patent specifications, in:

UK Cl (Ed.P): F2C

Int Cl (Ed.6): F16H 45/02

Other:

Documents considered to be relevant:

Category	Identity of document and relevant passage	Relevant to claims
X, P	GB 2280733 A (LuK) See Fig. 1	1
X, P	GB 2275513 A (LuK) See Fig. 11.	1 - 7 at least
X	GB 2127916 A (DAIMLER) See the Fig.	1

X	Document indicating lack of novelty or inventive step	A	Document indicating technological background and/or state of the art.
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